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SAND AND GRAVEL EXCAVATION;

PART 5: MOTOR-TRUCK HAULAGE, CONVEYOR-BELT HAULAGE, PUMPS AND PIPE LINES, BARGES AND TOWBOATS, AERIAL TRAMS

BY

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SAND AND GRAVEL EXCAVATION - PART 5: MOTOR-TRUCK HAULAGE, CONVEYOR*BELT HAULAGE, PUMPS AND PIPE LINES, BARGES AND TOWBOATS, AERIAL TRAMS[1]

By J. R. Thoenen[2]

INTRODUCTION

This circular is part 5 of the third paper (entitled "Excavation") of a series summarizing the technical problems involved in the production and preparation of sand and gravel. Part 1 discussed the use of power shovels, draglines, and excavator cranes; part 2, power scrapers, slackline cableway excavators, and hydraulic monitors; part 3, hydraulic, clamshell, ladder, and dipper dredges; and part 4, haulage systems in general, with particular reference to locomotive and car, hoist and car, and remote-control systems. This circular discusses the use of motor trucks, conveyor belts, pumps and pipe lines, barges and towboats, and aerial trams.

MOTOR-TRUCK HAULAGE

During recent years motor trucks have gained considerable ground as a successful competitor of locomotive haulage for quarries and gravel pits. Often where trucks have displaced locomotives and cars the operators readily admit an increase in both labor and total haulage operating costs but point to some other advantage as offsetting the extra expense. In other instances haulage costs have been reduced by the use of trucks. In still others operators, having replaced locomotive haul with trucks, have again returned to the earlier method. Obviously gasoline-driven motor trucks cannot successfully compete with other types of haulage under all conditions. The apparent' growth in the number of advocates of truck haulage, however, indicates that it has passed the experimental stage and become a permanent competitor to other types.

Probably the basic reason usually advanced for the increased use of trucks is summed up in the one word "flexibility."

Motor trucks require no rails and ties, although for most economical use it has been found best to provide a well-maintained roadway for them. However, for temporary haulage, trucks can usually service excavating machines without previous road preparation other than the clearing of obstacles and filling of holes. Locomotive haulage requires the laying of tracks and at least some ballasting for even temporary operation. Trucks have more power per ton of load, accelerate quicker, and are capable of shorter time cycles on short hauls. On long hauls the locomotive system will usually handle a larger tonnage per unit of labor and time, even though the speed may be slower. On hauls of several miles over well-ballasted track, however, locomotives may equal or surpass truck speed. Trucks can negotiate steeper grades and sharper curves than locomotives, and these items are of paramount importance in many pits.

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[1] The Bureau of Mines will welcome reprinting of this paper, provided the following footnote acknowledgment is used: "Reprinted from U. S. Bureau of Mines Information Circular 6875."

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Adequate motor-truck equipment operated under good management can supply continuous service to excavator units, and as this type of haulage uses a larger number of independent units delays at the excavator due to the haulage system are liable to be briefer.

One advantage of trucks over other types of haulage is that they can be diverted from pit usage to delivery equipment from plant to consumer when required. Frequently a gravel-pit operator is called upon to fill an order for bank-run material. If his pit is equipped with motor-truck haulage he can usually avoid rehandling material at the plant. Trucks can be loaded by the excavator and sent direct to the consuming point. With other types of pit haulage, bank-run material must usually be unloaded at the plant and reloaded to the delivery equipment.

Trucks require more labor per ton hauled than locomotives. They may consume as much or more fuel per unit handled, and because of the larger number of motor units and the use of rubber instead of steel treads repair and maintenance costs may be greater. Ordinarily, however, for equal capacity the initial investment is lower for truck haul.

Often locomotives may be operated in wet weather when trucks cannot function owing to slippery roads. If a well-maintained roadway has been built between pit and dump this disadvantage largely disappears.

For pits producing large tonnages (several thousand tons daily) locomotives with their greater unit capacity have the advantage over truck haulage because less labor is required for the lower number of haulage units and there is less traffic interference within the pit.

Both types of haulage require careful, adequate, and continuous supervision and maintenance to continue efficient.

Structural Limitations

Motor trucks are built in so many types and sizes that space does not permit enumeration of their details. The drive unit and chassis are usually standardized by each manufacturer, although the standards of one builder may vary greatly from those of another. Bodies may be built and supplied by the truck builder, but in many instances they are manufactured by separate concerns or by the gravel producer himself. The correlation of body size to tire load is important, and if truck bodies are furnished by concerns other than the truck builder care should be exercised that the load capacity does not exceed the designed tire loading as excessive maintenance costs may then result. Bodies range from 1 to 15 or more tons in capacity and may be arranged to dump to one or both sides or to the rear end. Some are designed to be dumped by gravity, others by mechanism operated by the truck motor, and still others by independent means. They may be constructed of wood or structural steel, and recently aluminum alloys have received considerable commendation by operators who have experimented with them in attempts to decrease the dead load and increase the pay load on the same unit.

Trucks may be operated as single units or may haul one or more trailers, each of which may have an equal or even greater capacity than the truck itself. The addition of trailers, however, is usually attended by a decrease in flexibility, as such units are slower and require extra maneuvering, both at the excavator and the dump; however, the use of one or more trailers has been found economical for long delivery hauls, particularly in western practice. Trailers have usually been discarded for pit to plant service.

Trucks are well-adapted for either short or long pit hauls. However, more trucks or larger cargo capacities are needed with longer hauls, hence initial expense is greater. There is considerable controversy among operators as to the most economical size of truck. Some maintain that a larger number of small trucks is better, while others prefer a smaller number of large trucks. In any event, the truck body should be of such size as to serve the shovel dipper or dragline bucket adequately and should in no case be less than the maximum capacity of one bucket or dipper. It would appear that for most economical operation the same rule should apply to truck bodies as to cars, namely, that the body should be large enough to hold at least three dipper or bucket loads. Such a rule would limit truck haulage to shovels or draglines with bucket capacities of 4 or 5 cubic yards or less. One probable reason for this limitation is the heavy impact the truck springs must absorb in receiving loads from larger excavators. Truck springs are more flexible than those used on railway equipment, and hence they are not designed for such heavy impact. Sometimes, specially designed springs have been used, but this seems to be a doubtful expedient as the impact is then delivered with less cushioning to the axles and bearings which on trucks are of less rugged construction than those on railway equipment.

Most trucks are equipped with the conventional 3 or 4 speeds and reverse, but in some the gear ratios are arranged to provide as many as 9 forward and 3 reverse speeds.

Trucks can be operated successfully on grades up to 15 percent and even steeper for short hauls. They can make nearly right-angled turns of small radius. The author has even observed rear-dump trucks backing from excavator

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to dump and moving forward for the empty return. This expedient is, of course, suitable only for hauls of a few hundred feet. The time required to turn around, however, is saved and the length of the haulage cycle is thereby reduced.

Probably the majority of trucks used for pit haulage dump to the rear. This requires the truck to consume valuable time in backing to the dump. Where trucks are equipped to dump to the side this loss of time may frequently be avoided. Side dumping, however, seems to offer greater structural difficulties, and this may account for its less frequent usage.

Pneumatic tires have largely displaced solid tires for both front and rear truck wheels. The drive is usually through the rear wheels, although some operators prefer both front- and rear-axle drive. To increase traction on rear-wheel drives some builders design each rear wheel to carry two tires. Others use four wheels on the same axle. Still others use two rear axles, either or both of which may be driven and may carry 2 or 4 wheels or tires. Wheel and tire design should be correlated to the weight-carrying capacity of the truck body, as most of the load is supported by the rear or drive wheels and tires.

The bodies of trucks are approximately the same size per ton of load and have the same clearance height above the ground as cars. The driver's cab usually extends above the body and being close to one end prevents a clear swing of the excavator bucket or dipper over the body, thus tending to increase the dumping time of the excavator. The bucket or dipper may be swung above the top of the cab, but when it is dumped the impact on the truck body and springs is then increased.

Trucks are built for speeds up to 40 or more miles per hour, although maximum speeds are seldom attained in pit haulage.

Service Equipment Required

Truck haulage systems to be efficient should have facilities for completely servicing and overhauling motors, tires, and dump bodies. This requires repair shops and garages of sufficient capacity to house all equipment. In cold weather trucks are frequently difficult to start, hence garages should be heated. Where heated garages are not available some operators have adopted the practice of completely draining the radiators at the end of the shift and providing a suitable supply of hot water for refilling them in the morning. This practice, however, is open to criticism owing to the tendency of the cylinder to crack under the strain of sudden expansion due to the introduction of hot water in a cold cylinder block.

While trucks can operate over fairly soft surfaces and even through loose sand, time and money are saved if hard-surfaced roads are prepared for them. Many operators have found it economical to build roads of standard highway construction between pit and plant, extending them as close to the excavator as possible. These roads should be kept in good repair and all holes promptly filled. At some pits graveled roads are built and a smooth surface is maintained by causing one or more trucks to drag behind them a grader made of railroad rail.

Trucks must be serviced regularly for fuel and oil, but this service is usually rendered at the garage during the noon hour or after the shift is finished.

Many operators insist upon a daily washing and mechanical inspection and periodic greasing of all trucks and bodies. At some plants the truck driver is held responsible for the mechanical care of his truck and paid accordingly. At others a chief mechanic is employed whose sole duty is to see that all trucks are properly serviced.

Recently a new type of truck accessory has been put to use.[3] This consists of a so-called "truck-skip," and is used principally for pits employing hand loading.

Capacity

The capacity of truck-haulage systems is largely a matter of organization and management. The very flexibility of truck units makes careful supervision and timing imperative in order that lost time be kept at a minimum. So far there have been few published analyses of truck haulage problems in the technical literature. Most of these have covered long hauls (several miles) from mine to treatment plant or railway siding and are not applicable to pit conditions. Andrew P. Anderson made an analytical study of truck haulage in connection with the construction of concrete highways, and while his study is not directly applicable to gravel-pit haulage much of the material contained in it can be so utilized.[4] Anderson found that on short hauls (less than 1 mile) the main factor in determining haulage capacity was not the road speed of the unit but the length of time the truck must spend while being loaded and dumped. He also found that, given equal road speeds and normal operating conditions, a 2-, 3-, or 4-batch truck does not deliver to the mixer at the pavement as many batches per hour as 2, 3, or 4 single-batch trucks would. Without further analysis this implies that smaller trucks have greater unit capacity than larger trucks, but when it is recalled that a 4-batch truck must discharge one batch to the mixer, then wait until the mixer is ready for another before dumping the second batch, the reason for this is clear in the lost time at the mixer.

From an analysis of Anderson's study of truck haulage in 122 paving jobs it appears that 9 to 10 percent of the total available haulage time was lost because of faulty equipment or faulty operation of the haulage units. This percentage is computed after deduction of all other time losses attributable to causes or equipment other than the haulage unit itself. Moreover, it does not include the time required to load and dump trucks. The 10-percent loss may then be assessed to factors avoidable with well-maintained trucks operated under perfect organization. Since it is not possible for any truck-haulage system to operate with 100-percent efficiency it is reasonable to assume that an average time loss of 10 percent due to the equipment and operation of any truck haulage system may be expected and that variations up or down will result from the degree of efficiency of equipment maintenance and operation.

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[3] Rock Products, Quarry Operator Invents Novel Truck-skip: June 1934, p. 51.
[4] Anderson, A. P., Truck Operation and Production in Concrete Paving Work: Public Roads, vol. 11, no 12, February 1931, p. 247.

As pointed out in previous parts of this paper no excavating unit can be expected to operate indefinitely at theoretical capacity, and the working capacities were computed by deducting various percentages from the theoretical. In actual operation excavation equipment does not function in a straight-line curve represented by a percentage of theoretical capacity but in a more or less irregular curve presenting numerous peaks and valleys, the average of which will approach the straight-line curve of working capacity. The capacity of any haulage unit will be found to follow the same rules, especially with such a flexible system as truck haulage. Then, if the peak of the excavator curve coincides with the peak of the haulage curve highest combined efficiency results. Conversely, when the valleys of both curves coincide lowest efficiency results. The haulage unit must be designed to serve the excavator at its peak operation and therefore will have excess units available during less productive periods. These excess units in the circuit must inevitably cause a certain amount of time loss while waiting to be loaded. The time lost in this way cannot be attributed directly to either the excavator or the truck and therefore must be added to the 10 percent assessed to the truck system itself. The amount of loss from this cause will vary from day to day, and even from moment to moment, on any job, depending upon local conditions. No published studies are available from which to evaluate this loss, but from the author's observations efficient all-around service may require the time lost to equal the time required to load one truck. On that basis, the second truck in line would arrive at the excavator just after the first one had started loading. The time consumed by each truck at the excavator would then be twice the actual loading time.

Frequently, time is also lost at the dump with truck haulage. This loss depends largely upon the accommodations provided for unloading. In some instances trucks are run over a pit or hopper and dumped without stopping, but ordinarily the truck comes to a complete stop and stands still until the load is discharged. With trucks designed for side dumping the delay at the dump is usually no more than the actual time required to dump. With rear-dumping trucks further time is usually lost in backing the truck to the dump. Trucks equipped with gravity-dump bodies are frequently difficult to dump owing to the maldistribution of the load by the excavator. The total time required to maneuver and dump the truck will depend entirely upon local conditions. Any truck in good condition should discharge a load of gravel in 20 seconds or less, and maneuvering to a dumping position should not require more than 40 seconds, making a maximum dumping time of 60 seconds. This issumption of the author is arbitrary but it is his opinion that the consumption of more

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than 60 seconds by a motor truck at the dump is cause for serious study.

Besides the waiting time at the excavator, there may be corresponding time lost at the dump. This is largely eliminated if the dump is so arranged that the truck can discharge its load without stopping. Where the dumping hopper is too small to accommodate more than one truckload at a time, delay caused by waiting is apt to be unduly large. With the best arrangements there is apt to be some waiting loss due to divergence or coincidence of peaks and valleys in the operating curves of trucks and dump mechanism. For calculation, however, the author feels that one period of loading time at the excavator should be ample to cover the average loss there and include the average loss from a similar cause at the dump, hence no delay factor is used at the dump other than the estimate for maneuvering and dumping.

The truck should, of course, spend the greatest portion of the available hauling time in actual travel between excavator and dump. The empty return distance may vary somewhat from the loaded haul, and the return speed will also vary from the loaded speed, therefore the round-trip speed should be computed by dividing the actual distance covered in the complete cycle by the actual time traveling.

The speed of trucks will be influenced by many factors, among which may be mentioned the mechanical condition of the truck itself; the grade, curvature, and surface condition of the roadway; the amount of traffic interference; whether all units are capable of maintaining the same speed; and the length of the haul. With respect to truck speed Anderson concludes from his study:

> First, that with good roads, proper equipment in good condition, freedom from traffic interference, and able management fairly high hauling speeds can be maintained rather consistently.
>
> Second, that with any one of these requirements lacking the uniform maintenance of high speeds is very difficult and probably impossible.
>
> Third, that a high speed is ordinarily of little productive value unless it can be maintained fairly consistently and includes all the hauling units.

Without doubt, these conclusions can be applied equally to pit haulage.

Anderson found that round-trip truck speeds ranged from 6 to 30 miles per hour and averaged 18 miles per hour. An analysis of his table comparing loaded speed with empty return speed shows that the return speed ranged from 4.5 to 38 percent faster than the loaded speed on normally loaded trucks and up to 89 percent faster when overloads were carried on light trucks.

He found further that, other conditions being equal, the average round-trip speed of trucks increased almost invariably with an increased hauling distance but not in computable ratio.

Calculation of the capacity of motor-truck haulage systems involves:

1. The loading speed of the excavator served.
2. The length of round-trip haul.
3. The average round-trip speed.
4. The time consumed in loading and dumping.
5. The time lost from other causes.
6. The cargo capacity of the truck unit.
7. The number of trucks.

From these elements the following formula is borrowed from Anderson's paper, but the factors represented by the components are changed to represent pit haulage.

$$N = \frac{L}{Smt} + \frac{T}{mt},$$

in which

N = number of trucks required;

L = round-trip length of haul, in feet;

S = average round-trip speed, in feet per minute;

T = total time used by each truck in loading, dumping, maneuvering, and waiting on each round trip, in minutes;

t = loading speed of the excavator, in minutes per ton at working capacity;

m = number of tons carried by each truck.

Example 9

Assume an operator has a 1-1/4 cubic yard shovel digging loose gravel which he wishes to serve with trucks. His round-trip haul is 1,500 feet, and road conditions are such that his trucks can average 15 miles per hour while running. It is further assumed that the trucks will lose 10 percent of their available hauling time for various causes and that they will require 1 minute to back and dump at the plant.

From table 9 in part 1 of this paper it is found that the maximum working capacity of 1-1/4-cubic yard shovel is 206 tons per hour.

The loading speed of the shovel is then

$$\frac{20}{206} = 0.291 \text{ minute per ton.}$$

Assuming the gravel weighs 3,000 pounds per cubic yard, each bucket will deliver 1.875 tons. A 5-cubic yard truck will carry 7.5 tons and will be loaded by 4 dipper loads. (Variation in dipper loading is accounted for in calculating shovel working capacity, hence each dipper is here calculated as full.)

The time required to load one truck will then be

$$0.291 \times 7.5 = 2.18 \text{ minutes.}$$

Since 15 miles per hour equals 1,320 feet per minute, the running time of the truck will then be

$$\frac{1500}{1320} = 1.136 \text{ minutes.}$$

The truck cycle, exclusive of lost time, will be as follows:

	Minutes
Loading	2.18
Waiting at shovel	2.18
Running	1.136
Dumping	1.00
Total	6.496

Since it was assumed that there would be a 10-percent loss of time for various causes this 6.496 minutes represents 90 percent of the actual cycle; hence the actual cycle equals

$$\frac{6.496}{0.9} = 7.218 \text{ minutes.}$$

The standstill time plus lost time of the truck per trip is then

$$7.218 - 1.136 = 6.082 \text{ minutes.}$$

Substituting in the formula,

$$N = \frac{L}{Smt} + \frac{T}{mt} = \frac{1500}{1320 \times 7.5 \times 0.291} + \frac{6.082}{7.5 \times 0.291} =$$

$$0.52 + 2.79 = 3.31 \text{ trucks.}$$

To check:

Minutes

Loading time ----------2.18

Waiting time ----------2.18

Running time ----------1.136

Dumping time ----------1.00

Lost time ------------- .722

Total cycle ---------7.218

$\frac{60}{7.218}$ = 8.31 trips per truck per hour;

8.31 x 7.5 = 62.3 tons per truck per hour; and

$\frac{206}{62.3}$ = 3.31 trucks required.

With such a short haul it is doubtful if the wait at the shovel need be as great as is assumed in this case, and probably 3 trucks would be sufficient, in continuous operation, especially as the maximum loading capacity of the shovel was used in the computation. However, trucks, like all other kinds of machinery, wear out and require repairs, hence an extra truck as standby equipment is indicated, making 4 trucks.

Example 10

Assume that pit conditions are the same as in example 6 under car and locomotive haulage in part 4 of this paper and that it is desired to substitute truck haulage for the locomotive haulage calculated there.

With the same haulage route the total round-trip haulage distance will be 8,300 feet. At an average truck speed of 15 miles per hour the speed per minute is 1,320 feet. The two 1-1/2 cubic yard shovels will have a combined maximum working capacity of 530 tons hourly and a minimum of 280 tons. In computing locomotive haulage, however, the average was used, or 400 tons hourly, so the same loading rate will be used here. The combined rate is then

$$\frac{60}{400} = 0.15 \text{ minute per ton.}$$

However, the two shovels cannot load the same truck at the same time, hence the loading rate per shovel must be used. Thus,

$$\frac{60}{200} = 0.3 \text{ minute per ton.}$$

It is assumed that each truck can haul a pay load of 6 cubic yards or 9 tons. Then,

	Minutes
Loading time for each truck, 9 x 0.3 ----------------	2.7
Waiting time ----------------------	2.7
Running time, $\frac{8,300}{1,320}$ ----------------	6.29
Dumping time ----------------------	1.00
Operating time --------------------	12.69
Lost time (1/9) -------------------	1.41
Total cycle -----------------------	14.10

The standstill plus lost time will then be

$$14.1 - 6.29 = 7.81 \text{ minutes.}$$

Substituting in the formula,

$$N = \frac{8300}{1320 \times 9 \times 0.3} + \frac{7.81}{9 \times 0.3} = 2.33 + 2.89 = 5.22 \text{ trucks}$$

required for each shovel or, say, 12 trucks in continuous operation.

To check:

Truck cycle = 14.1 minutes;

$$\frac{60}{14.1} = 4.25 \text{ trips per hour per truck;}$$

$$4.25 \times 9 = 38.3 \text{ tons per hour per truck;}$$

$$\frac{400}{38.3} = 10.44 \text{ trucks required.}$$

If 12 trucks are used the capacity of the haulage unit would be 2,760 tons in 6 hours or 360 tons more than the 2,400 required.

CONVEYOR-BELT HAULAGE

Belts have been used for years as a means of conveyance between plant units within a single structure, between plant buildings, or between other fixed points. The introduction of the tripper and the shuttle conveyor permitted collection from, or discharge to, more than one point by a single belt. The various points, however, were invariably in a straight line coinciding with the axis of the belt. Application of sectionalized construction to conveyors increased flexibility but still confined operation to straight lines. The belt conveyor was considered permanent equipment, although it was capable of extension by the various devices cited. With the introduction of light-weight, easily handled sections the belt conveyor forced its recognition as a potential competitor for other types of pit haulage. It has been so used by a number of gravel-pit operators with successful results.

The field conveyor is simply an ordinary conveyor redesigned along semi-portable lines and adapted to field use. The load and return sections of the belt are carried on rigid structural-steel or alloy frames built in unit sections designed for easy longitudinal extension or lateral movement. The belt itself is in easily connected lengths to fit the added sections.

Field conveyors must still operate in straight lines, but flexibility is provided by sectionalized units. They have been adapted to field haulage in various ways and to fulfilling various functions. They may constitute the sole connecting link between excavator and plant. In so doing they may receive material from the excavator and deliver it direct to the treatment plant either above, below, or level with the pit floor. The conveyors must be of light weight and sectionalized to provide facility of movement in following the progress of the excavator. The plan of excavation may require the excavator to follow a circular or elliptical path or to move in a straight line away from the plant. With a curved working face the conveyor must be constructed in a single unit with one end stationary at the dump and the other moving radially with the excavator, or it must comprise two or more sections working together.

Any pit with a curved working face depending upon a single unit radial conveyor for haulage must soon cover such an extended area that it is uneconomical to move the entire conveyor for each move of the excavator. If the plan of excavation extends in a straight line from the plant it is necessary merely to add successive sections to the conveyor to follow the excavator. However, even with such a plan a distance will finally be reached where it is more economical to displace most of the sectionalized conveyor with permanent construction. With either excavation plan a point is reached at which part of the pit conveyor must be of permanent construction served by a shorter movable unit.

The short movable unit may be a sectionalized conveyor, or it may be an entirely different type of haulage system, such as a fleet of motor trucks or cars and locomotives, or even a pump and pipe line. The belt conveyor then becomes a main haulageway fed by one or more flexible units.

The sectionalized pit conveyor may be so arranged that it follows the excavator, taking material from it and delivering it to another type of main haulage. It then becomes the flexible feeder for a permanent haulage system, such as an aerial tramway, or even a car and locomotive system.

The field conveyor may also be a permanent structure between two fixed points in the pit; at one point material is delivered to it from the excavator by a more flexible type of haulage and at the other the conveyor discharges to a third type of haulage which in turn delivers to the plant. Such a scheme might include truck haulage from the shovel to a centrally located conveyor which delivers to an engine plane for hoisting or lowering to the plant.

While the pit conveyor is restricted to straight-line delivery it is thus capable of considerable flexibility when used with other haulage systems.

Pit conveyors are restricted to use in dry pits or bank deposits. They are not suitable for haulage from wet pits unless some dewatering device is used. They are capable of handling small or large tonnages and, when properly installed, of delivering a continuous flow of material to the plant which is advantageous to efficient plant operation. They require comparatively little power, and as they are fully mechanized are practically automatic in their operation. They require maintenance labor only, and when they are properly installed and not overloaded operating costs are low. Initial and installation costs may be comparatively high, and the cost of moving and realining a long conveyor is expensive.

Structural Limitations

Belt conveyors are endless belts running over supporting idlers and driven through arc contact with one or more driven pulleys. The load is conveyed on the belt itself, hence the belt must be horizontal in cross-section if flat, or if troughed, the edges must be at the same elevation, otherwise the load will spill off the low side. In this position the belt is flexible vertically but not horizontally, hence while it can negotiate vertical curves it cannot operate around horizontal curves. Therefore belt conveyors can carry loads on the level or up or down inclines, but they cannot deviate horizontally from a straight line between terminals.

The belt may be flat or troughed in cross-section, but, as a flat belt offers little resistance to spillage for particles of the load heaped on the center of the belt, troughed cross-sections are preferred for sand and gravel transportation. Troughed belts have about twice the capacity of flat belts of equal width and speed. The shape of the trough is governed by the position of the idler rolls supporting the belt and varies with the experience of manufacturers and operators. The supporting idlers are constructed so that one or more level rolls carry the center of the belt, and the edges are raised to form the trough by means of one or more troughing rolls on each side which rotate around axles at an angle above the horizontal. The total troughing angle seldom exceeds 30°. The full curvature may be accomplished in a single angle on each side, in which case the angle is usually 20°, or partly in one roll set at a low angle and the balance by second or third rolls at higher angles. The usual arrangement is 15° on the first and 15° more on the second. The details of idler design are highly perfected, with many variations suited to the handling of different classes of material. If the troughing angle is too sharp, excessive stress and wear are put on the belt and it tends to crack at the line of bend; therefore, troughed belts must be thick enough to prevent cracking and may require more plies than flat belts. However, this is seldom the case; for example, a flat belt must be thick enough to prevent excessive sag between idlers. The design must be governed by local conditions. Table 58 gives the minimum and maximum plies required for troughed belts.

TABLE 58.- Minimum and maximum plies for troughed belts

Belt width, inches	Minimum plies required to support load			Maximum plies for troughing
	Sand	Gravel	Boulders	
12	3	4		4
14	3	5		5
16	3	5		5
18	4	5	6	5
20	4	5	6	6
24	4	6	7	6
30	5	6	7	8
36	5	6	8	9
42	6	7	8	10
48	6	7	9	12
54	7	8	10	13
60	7	8	10	14

While a certain amount of vertical curvature is permissible in a conveyor belt it should be kept at a minimum and permitted only through arcs of large radius. Conveyor belts when running are under a tension which varies with the load. If a belt is running horizontally at the loading point and must then ascend a grade the change from the level to the inclined section must be a curve of such radius that the weight of the empty belt holds it on the rolls in spite of the tension required. The maximum adverse conditions requiring curves of the longest radius occur with conveyors having a profile in which there is a long level section carrying a

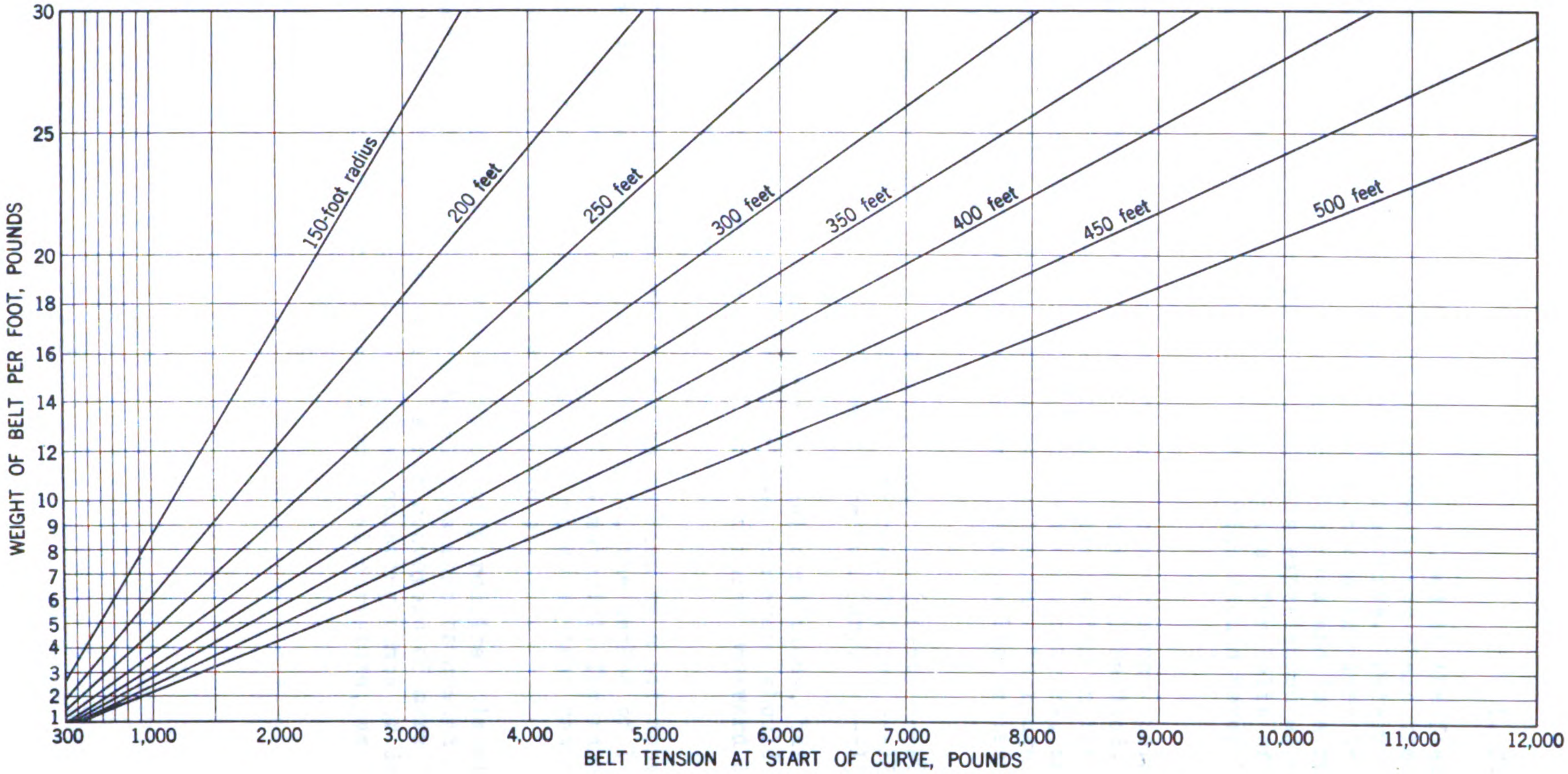

Figure 19.—Chart showing radius for vertical curves in conveyor belts for various belt weights and tensions.

heavy load, followed by a steep incline. The heavy load on the flat portion of the belt causes heavy tension, and if the curve is too sharp the empty belt will be lifted off the troughing idlers at the curve. Determination of the exact curvature required for specific conditions involves the use of complicated mathematical solutions of catenary curves and is a function of conveyor design that should be left to engineers experienced along that line. Figure 19 shows the approximate radius required for belts of various weights operating under various tensions. The radius should never be less than 150 feet.

The maximum angle of inclination over which a conveyor may operate successfully depends upon the angle of repose of the material handled. It will vary with different kinds and sizes of particles making up the load, being greater for fine powders, cubical or rough-edged particles, and damp material and less for lumps, rounded particles, and either thoroughly dry or very wet material. The following are maximum permissible angles of interest to the sand and gravel producer.

Sand (damp)-----20°
Sand (dry) -----15°
Gravel, bank run (moist) ---20°
Gravel, unscreened (dry) ---18°
Gravel, screened (dry) ---15°

Steeper angles have been employed under unusual conditions, but those given should not be exceeded ordinarily. Inclined conveyors should be provided with some means of preventing the belt from running backward if accidentally stopped while under load.

The size of individual pieces which can be carried by a pit conveyor depends upon the width and inclination of the belt, the method of loading, and whether lumps are screened or mixed with fine material. The range of maximum-sized lumps screened or mixed with finer material that can be carried for various widths of belt is shown in table 59.

Field conveyors which must be shifted frequently require a level or smoothly inclined pit floor, otherwise inequalities must be graded or bridged to provide a smooth running belt. Field conveyors of more or less permanent construction can be installed to span existing humps or depressions in the floor which do not require excessive vertical curvature, as discussed previously.

TABLE 59.- Recommended maximum size of lumps for belt-conveyor haulage

Belt width, inches	Lumps uniformly sized, no fines, inches	Lumps in unsized feed, inches
12	1.5 - 2.0	2.0 - 4.0
14	2.0 - 2.5	2.5 - 4.0
16	2.5 - 3.0	3 - 5
18	3.0 - 4.0	4 - 6
20	4.0 - 4.5	5 - 7
24	4.5 - 5.0	6 - 10
30	6 - 7	10 - 14
36	7 - 9	12 - 18
42	8 - 11	14 - 20
48	10 - 14	16 - 24
54	11 - 15	20 - 28
60	12 - 16	24 - 30

The spacing of idler rolls depends upon the weight per cubic foot of the material carried and the width of the belt. Table 60 shows the recommended range.

TABLE 60.- Spacing for carrying rolls, feet[1]

Width of belt, inches	Weight of material, pounds per cubic foot			
	10 to 40	40 to 75	75 to 120	120 to 150
12	5.5	5.0 - 5.5	5.0	4.5 - 5.0
14	5.5	5.0 - 5.5	5.0	4.5 - 4.75
16	5.5	5.0 - 5.5	5.0	4.5 - 4.75
18	5.0 - 5.5	4.5 - 5.5	4.5 - 5.0	4.0 - 4.75
20	5.0 - 5.5	4.5 - 5.5	4.5 - 5.0	4.0 - 4.75
24	4.5 - 5.5	4.5 - 5.5	4.0 - 5.0	3.5 - 4.75
30	4.0 - 5.0	4.0 - 5.0	3.5 - 4.5	3.5 - 4.25
36	4.0 - 5.0	3.5 - 5.0	3.5 - 4.5	3.0 - 4.25
42	4.0 - 4.5	3.5 - 4.5	3.5 - 4.0	3.0 - 3.75
48	3.5 - 4.0	3.5 - 4.0	3.0 - 3.5	3.0 - 3.5
54	3.5 - 4.0	3.5 - 4.0	2.75 - 3.5	2.5 - 3.5
60	3.5 - 4.0	3.5 - 4.0	2.75 - 3.5	2.0 - 3.5

[1] Return rolls can be 8 to 10 feet apart.

Conveyor belts are built up of a number of thicknesses or "plies" of canvas or "duck", impregnated and covered with rubber. The maximum length of the conveyor depends upon the strength of the belt or its resistance to tension, which in turn depends on the number of plies used in its construction and the weight of the duck. The weight of the duck is calibrated by the weight, in ounces per yard, of material 42 inches wide. Canvas of this width commonly used for conveyor-belt construction weighs 28, 32, and 36 ounces per yard; 42-ounce duck is sometimes used for extra-heavy belt. The permissible working stress for each ply per inch of belt width, as recommended by manufacturers, is given in table 61.

TABLE 61.- Maximum working stress per ply, per inch of belt width

Weight of duck, ounces	Working stress, pounds[1]
28	20 - 25
32	23 - 28
36	25 - 30
42	35 - 40

---

[1] The reason for tabulating a range of working stress is due to its application. The stress applied by counterweights can be definitely controlled and, hence, kept to a minimum. If screw take-ups are used it is difficult to measure and control the stress, hence stronger belts are needed.

The tensional strength of a conveyor belt is therefore proportional to the number of plies of duck used in its construction. This strength cannot be built up indefinitely, however, simply by the addition of extra plies without other complications being encountered. Additional thickness stiffens a belt and reduces its flexibility for troughing and bending around drive pulleys. The maximum plies permissible for proper troughing flexibility were given in table 58. Table 62 gives the minimum recommended pulley diameters for various numbers of plies.

TABLE 62.- Minimum pulley diameters, in inches

Number of plies	Head and drive pulley	Tail, take-up, and snub pulleys[1]
3	15	10
4	20	12
5	24	16
6	30	18
7	36	20
8	42	24
9	48	28
10	54	30
11	54	36
12	60	36
13	66	42
14	72	42

[1] Bend pulleys may be 8 to 16 inches in diameter for any number of plies because of the small arc of contact.

The number of plies contained in standard conveyor belts, as listed in manufacturers' catalogs, is given in table 63.

TABLE 63.- Number of plies of duck in standard conveyor belts

Width of belt, inches	Plies		Width of belt, inches	Plies	
	Minimum	Maximum		Minimum	Maximum
12	3	4	30	5	8
14	3	5	36	5	9
16	3	5	42	6	10
18	4	6	48	6	11
20	4	6	54	7	13
24	4	7	60	7	14

The speed at which a conveyor belt travels should depend upon the load it carries. Generally, it should be as slow as possible consistent with uniform maximum loading. If a belt is underloaded the load will be carried in the middle, and the center will wear out faster. If the load carried by a belt already installed is less than its capacity at the operating speed used the speed should be reduced. Recommended maximum speeds are given in table 64.

TABLE 64.- Maximum recommended speeds, feet per minute

Belt width, inches	Speed of general-purpose belts	Speed of belts for abrasive materials
12	300	250
14	300	250
16	300	250
18	350 - 400	300
20	350 - 400	300
24	400 - 500	350
30	450 - 500	350
36 - 42 - 48	500 - 600	400
54 - 60	600	400

Conveyor belts of the same specifications made by various manufacturers will vary somewhat in unit weight, but the difference will be small. The average unit weight per ply, exclusive of rubber covers, may be calculated as follows.

	Weight per ply per inch of width per linear foot			
Duck	28 oz.	32 oz.	36 oz.	42 oz.
Pounds	0.021	0.024	0.027	0.032

The rubber cover on the load-carrying side of the belt ranges in thickness by sixteenths from 1/16 to 1/4 inch. That on the pulley side is usually not over 1/32 inch thick but may be 1/16 or even 1/8 inch for severe service. The weight of the rubber covering may be calculated as 0.018 pound per 1/32 inch of thickness per inch of width per linear foot.

A 5-ply, 30-inch conveyor belt of 28-ounce duck having a 1/8-inch load cover and a 1/32-inch pulley cover would then weigh approximately

		Pounds per foot
5 x 30 x .021	=	3.15
5 x 30 x .018	=	2.70
		5.85

Service Equipment Required

The only way in which the load carried can cause wear on a conveyor belt is through abrasion, and abrasion requires motion. If an object carried by a conveyor does not change its position with respect to the belt while in transit it causes no wear. The field conveyor should be designed to carry its load with the least change of load position while in transit. For this reason the method of loading a conveyor belt has a direct bearing upon the life of the belt. Field conveyors should never be loaded directly by mechanical excavators. When they are so loaded the gravel is moving vertically downward when it strikes the belt. The belt must convert this downward motion to one coinciding with its own line of travel. This requires time, during which the gravel is changing its position on the belt and causing abrasion. The motion due to impact also causes abrasion. In addition, mechanical excavators deliver material in batches which, if loaded directly, would cause concentrated loads at irregular intervals on the belt, resulting in a constant shifting of position of the load as it passed over each idler and excessive stresses in the belt. To obviate intermittent loading a portable hopper is usually set up over the belt into which the excavator drops its load. It may be simply a wooden or steel box with sloping sides, terminating in a chute over the conveyor. Such a hopper is usually an independent unit and is picked up and moved ahead by the excavator. The hopper may be part and built integrally with the end section of the conveyor. The type used ordinarily depends upon local conditions. Regardless of type, the design should be such as will convert the batch loading of the excavator to a regular and continuous feed to the belt.

The feed to the belt although continuous may be irregular if uncontrolled, hence the feed from the hopper must be controlled at all times. If the gravel is dry, free from lumps of clay, and graded in size so as to permit it to run freely then the simplest type of control is a chute closed by a sliding gate operated by rack and pinion. The gate can then be set at an opening that will permit the passage of gravel at the required rate to the belt.

If the gravel is not free-running, owing to contained moisture, clay, or boulders, the gate-controlled chute is not a very satisfactory feeder because of the tendency of the material to stick in the opening. If the narrowest part of the chute can be made three times the size of the average boulders little trouble should be experienced from choking. With sticky clay the size of the opening makes little difference, as the clay will stick and cause trouble anyway. Often, a small stream of water played on the chute facilitates movement of sticky gravel.

If the gravel is free from clay but contains considerable coarse material or boulders it is a good plan to construct the bottom of the chute of wire screen or grizzly bars. This permits the sand and fine gravel to drop through ahead of the coarse material and form a cushion on the belt, thus reducing the impact when boulders are received. Screens or grizzlies are not very satisfactory if the gravel contains clay because of its tendency to stop the openings.

Where chutes and gates do not afford good feed control some form of mechanical feeder should be installed. There are several types available, among which are the apron feeder, the reciprocating feeder, and the rotary or barrel feeder. The apron feeder (fig. 20) may be a short length of conveyor belt taking material from the chute and delivering it to the conveyor, or it may be built of steel with interlocking pans. The reciprocating feeder (fig. 21) is driven by one or more eccentrics. The carrying floor of the feeder may be horizontal or on an angle, as shown. It may be built of a single flat plate or screen, or it may be made of grizzly bars, each of which is set differently on the eccentric. If a screen or grizzly is used the conveyor extends back, as shown by dotted lines, and the fines fall through the feeder to form a cushion for the boulders. The roll or barrel feeder (fig. 22) is simply a cast-iron or steel cylinder revolving below the chute opening. All feeders obtain their material from an adjustable opening in the hopper bottom. Skirt boards should always be provided to prevent spillage as the gravel falls to the belt.

Conveyor idlers should never be placed where the falling material strikes the belt. By directing the loading to a point between rolls the impact is absorbed by the flexibility of the belt, and less abrasion occurs.

Irrespective of the type of feeder used the feed should be directed so that it moves in the same direction as the belt and at about the same speed. This is not always possible, but the closer the delivery approaches these ideal conditions the less abrasion there will be due to impact and subsequent movement on the belt.

Capacity

The capacity of a field conveyor depends upon the method of loading, the cross-sectional area of the load, the width and speed of the belt, and the weight per cubic foot of the material handled. If the loading method is inadequate to furnish the maximum cross-sectional area the belt can carry, or does not supply a uniform flow of material to the belt, the capacity is

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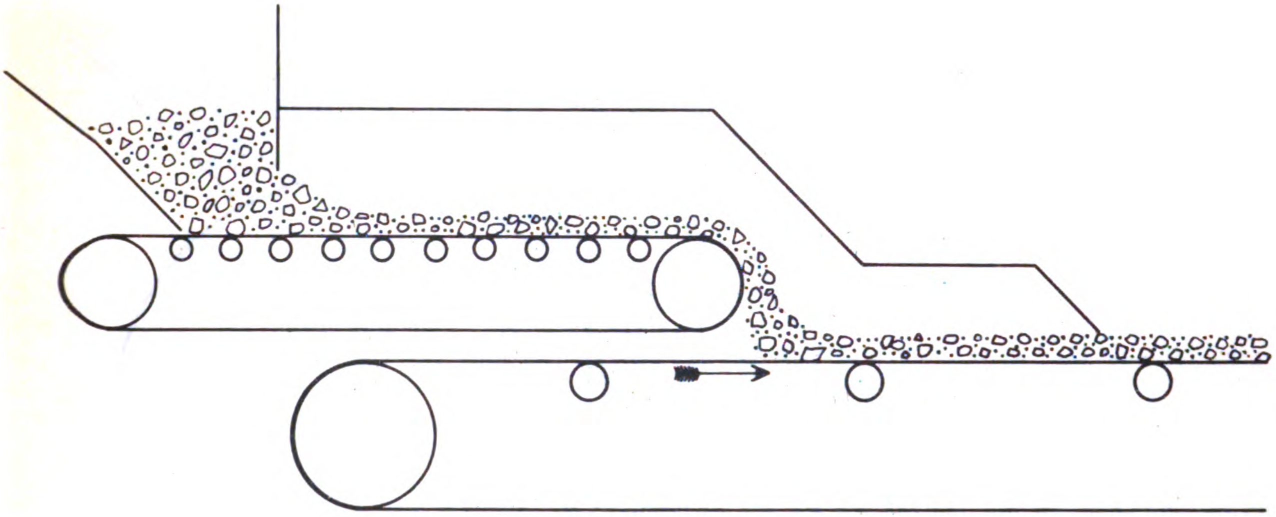

Figure 20.— Apron feeder.

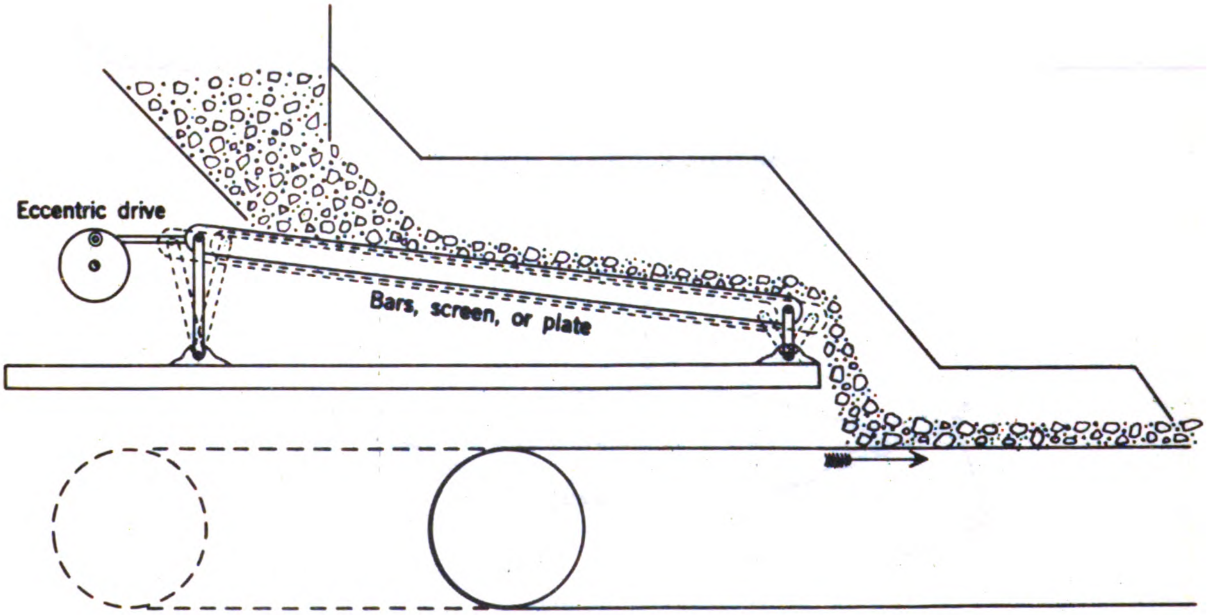

Figure 21.— Reciprocating feeder.

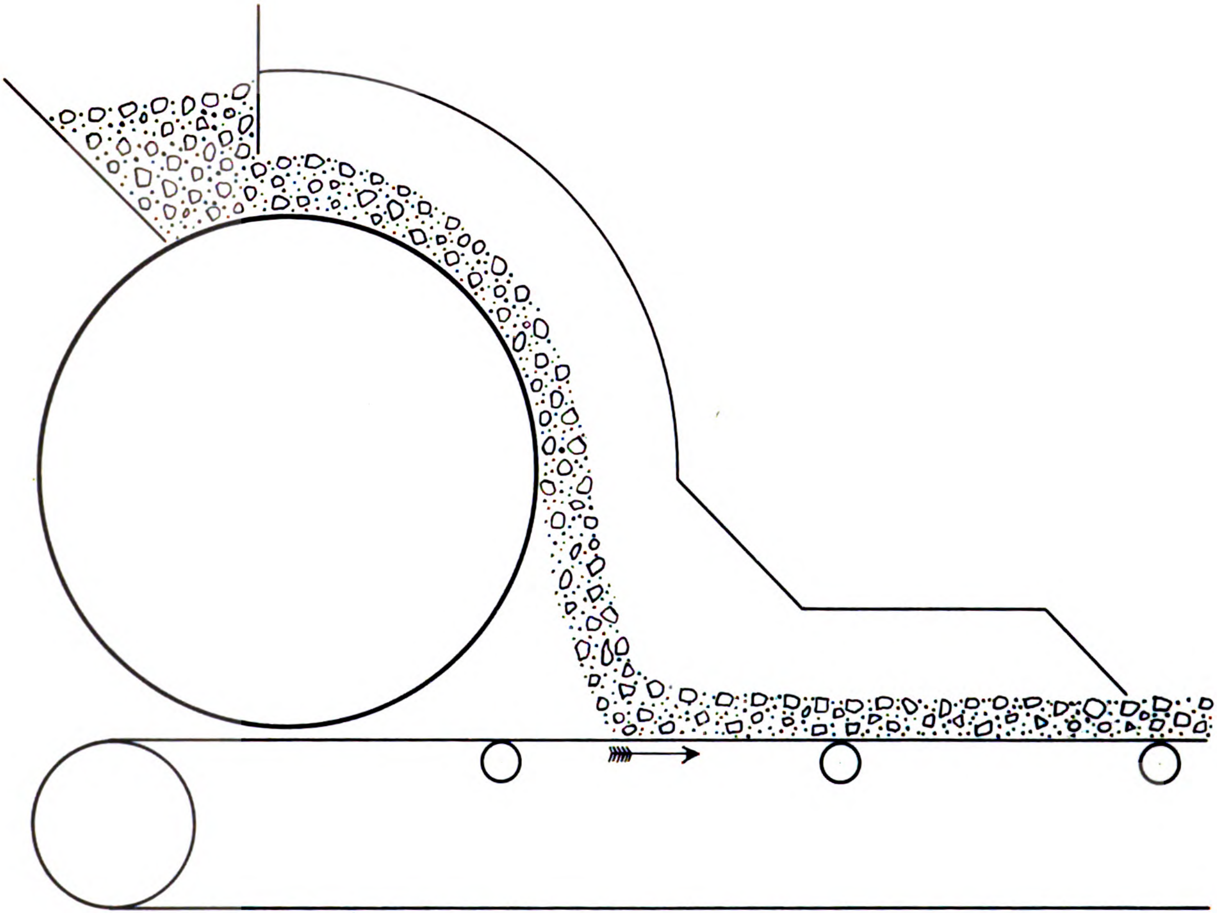

Figure 22.— Roll or barrel feeder.

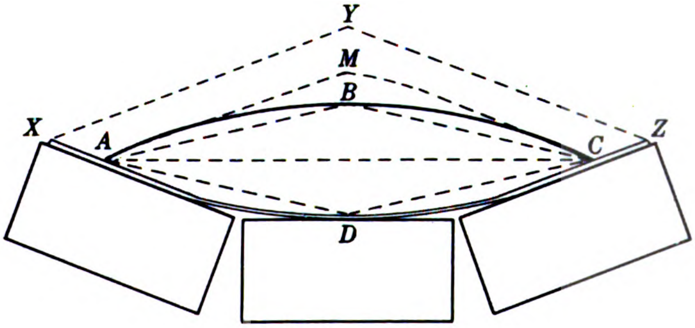

Figure 23.— Cross-sectional area of load.

correspondingly reduced. Therefore, the method of loading requires as careful consideration in calculating capacity as in calculating its effect upon the life of the belt. If the loading device cannot supply the maximum load which the belt width and speed allow, then the speed of the conveyor should be reduced. In the following discussion of conveyor capacities it is assumed that the design of the loading device is adequate for maximum belt capacity.

The trough or cross-sectional shape of the belt has a direct bearing on its haulage capacity. Flat belts have only about half the capacity of properly troughed belts. The proper trough shape depends upon the width and construction of the belt, the type of material carried, and the size, shape, and gradation of the individual particles. Hence, proper troughing depends upon the factors involved in each belt installation.

A field conveyor handling bank-run sand and gravel carries most of the load on the center of the belt. Theoretically, a belt can carry a load the cross-section of which coincides with the shape of the belt on the bottom and slopes upward and inward from each edge on an angle equivalent to the angle of repose of the gravel. Practically, the load never extends to the extreme edge of the belt, except when it is overloaded, and the upper part of the cross-section has a rounded perimeter due to settling as the belt passes over the idlers. The maximum cross-sectional area will thus depend upon the physical characteristics of the load itself, as well as upon the shape of the trough. For this reason the capacity of a belt may change from day to day as the physical characteristics of the gravel change in the working face of the pit. Wet weather may increase the moisture content of the gravel and cause it to build up higher on the belt. If the gravel is too wet this effect may be reversed, the load slumping to the edges. A change in the ratio of coarse to fine material in the feed is also reflected in the shape of the load.

Figure 23 shows the cross-section of a loaded conveyor belt. The area XYZD represents the theoretical load, impossible to reach because of settling and spillage. Area AMCD represents the section when first placed on the belt; this settles to area ABCD. The lateral angles will of course vary for different classes of material. A common method of calculating the average cross-sectional area of the load is to let the quadrangle ABCD (dotted lines) represent that area. The length AC is assumed as 0.8 the width of the belt and BD as 0.2 the width. If W is the width of the belt, the area is

$$\frac{0.8W \times 0.2W}{2} = 0.08W^2.$$

The maximum capacity, in cubic feet of material, may then be computed by the following formula.

$$L = \frac{0.08W^2 \times S \times 60}{144}$$

in which

L = the cubic feet of load per hour;
W = the width of the belt, in inches;
S = the speed of the belt, in feet per minute.

At a speed of 100 feet per minute the maximum capacity of the conveyor is then

$$L = 3\ 1/3\ W^2 \text{ cubic feet per hour.}$$

This formula is predicated upon the assumption that the cross-sectional area of the load varies directly with the square of the width of the belt. However, experiments have shown that with the same type of material the load can be spread relatively closer to the edges on a wide than on a narrow belt. Therefore, the square of the belt width must be multiplied by a factor which varies with the width of the belt. This factor has been shown empirically to range from 3.2 for a 12-inch belt to 4.0 for a 60-inch belt, assuming a belt speed of 100 feet per minute.

If this factor is designated as K, table 65 gives its value for belts from 12 to 60 inches wide in increments of 2 inches.

TABLE 65.- Values of K for various belt widths

Belt width, inches	K	Belt width, inches	K	Belt width, inches	K	Belt width, inches	K	Belt width, inches	K
12	3.2	22	3.37	32	3.53	42	3.70	52	3.87
14	3.23	24	3.40	34	3.57	44	3.73	54	3.90
16	3.27	26	3.43	36	3.60	46	3.77	56	3.93
18	3.30	28	3.47	38	3.63	48	3.80	58	3.97
20	3.33	30	3.50	40	3.67	50	3.83	60	4.00

The maximum belt capacity may be obtained from the following formula,

$$C = \frac{K \times W^2 \times S \times M}{200{,}000},$$

in which

C = the capacity, in tons per hour;
K = the factor of load cross-section;
W = the width of the belt, in inches;
S = the speed of the belt, in feet per minute;
M = the weight of the load per cubic foot, in pounds

Sand and gravel ranges in weight from 2,900 to 3,240 pounds per cubic yard, (107.4 to 120 pounds per cubic foot). Table 66 gives the range in maximum capacity, in tons per hour, for belts of various widths operating at a speed of 100 feet per minute and carrying materials of different weights per cubic foot. Table 66, A, gives the maximum capacity of belts of various widths handling sand and gravel at various speeds.

TABLE 66.- Maximum conveyor-belt capacities, in tons per hour at belt speed of 100 feet per minute

Belt width, inches	Factor K	Weight of material, pounds per cubic foot				
		100	107.4	110	115	120
12	3.2	23.0	24.7	25.3	26.5	27.6
14	3.23	31.6	34.0	34.8	36.4	38.0
16	3.27	41.9	45.0	46.0	48.1	50.2
18	3.30	53.4	57.4	58.8	61.5	64.2
20	3.33	66.6	71.5	73.3	76.6	80.0
24	3.40	93.0	105	108	112	118
30	3.50	157	169	173	181	189
36	3.60	233	251	257	268	280
42	3.70	327	350	359	375	392
48	3.80	438	470	482	504	525
54	3.90	568	610	626	654	682
60	4.00	720	774	792	828	864

TABLE 66, A. - Maximum average capacities of belt conveyors, tons per hour[1/]
(Sand and Gravel)

Belt width, inches	Belt speed, feet per minute (gravel ranging from 107.4 to 120 pounds per cubic foot)											
	100		150		200		250		300		350	
	From	To	From	To	From	To	From	To	From	To	From	To
12	24.7	27.6	37.1	41.5	49.5	55.3	61.9	69.2	74.3	83.0		
14	34.0	38.0	51.0	57.0	68.0	76.0	85.0	95.0	102	114		
16	45.0	50.2	67.4	75.3	90.0	100	112	125	135	151		
18	57.4	64.2	86.1	96.2	115	128	143	160	172	192	201	224
20	71.5	80	107	120	143	160	179	200	215	240	250	280
24	105	118	158	176	210	235	263	294	316	352	368	411
30	169	189	254	283	338	378	423	472	507	567	592	662
36	251	280	376	420	500	560	627	700	752	840	878	980
42	350	392	526	588	702	784	877	980	1,050	1,175	1,225	1,370
48	470	525	706	788	940	1,050	1,175	1,310	1,410	1,575	1,645	1,840
54	610	682	916	1,020	1,220	1,360	1,525	1,700	1,830	2,050	2,140	2,390
60	774	864	1,160	1,300	1,550	1,730	1,930	2,160	2,320	2,590	2,710	3,025

Belt width, inches	Belt speed, feet per minute (gravel ranging from 107.4 to 120 pounds per cubic foot)									
	400		450		500		550		600	
	From	To	From	To	From	To	From	To	From	To
12										
14										
16										
18	230	257								
20	286	320								
24	420	470	473	529	526	588				
30	676	756	762	851	846	945				
36	1,000	1,120	1,127	1,260	1,250	1,400	1,380	1,540	1,500	1,680
42	1,390	1,565	1,575	1,760	1,750	1,960	1,920	2,150	2,100	2,350
48	1,880	2,100	2,120	2,360	2,350	2,630	2,590	2,890	2,820	3,150
54	2,440	2,730	2,750	3,070	3,050	3,410	3,360	3,750	3,660	4,100
60	3,090	3,460	3,480	3,890	3,860	4,320	4,250	4,750	4,640	5,180

[1/] Working capacities probably do not exceed 60 to 75 percent of these quantities.

The capacities shown in the tables are based on the following assumptions: (1) The loading facilities are adequate to deliver the load to the belt at its maximum carrying rate; (2) the size gradation of the gravel is suitable for the width, speed, and inclination of the belt; (3) the applied power is ample; (4) the conveyor construction is such as to give the proper trough shape for the gravel handled; and (5) the belt has sufficient strength to withstand the stresses set up.

It must also be remembered that the capacities in table 66 are the average maximum for a belt speed of 100 feet per minute. For belts running at greater speeds the figures should be multiplied by the speed in hundreds of feet per minute.

The average working capacities will always be less than the tabulated figures because of imperfect loading and delays.

If the gravel contains a large percentage of boulders, their size fixes the width of the belt (see table 59), and the speed should be as slow as possible to give the required capacity.

With a high ratio of sand and fine gravel the capacity fixes the belt width, but here again the speed should be as slow as possible.

A slow-speed belt will carry a deeper load than a high-speed one.

Average working capacities probably will not exceed 75 percent of the calculated maximum rate.

Power

The power used to operate a conveyor belt may be taken from any type of prime mover, such as a steam or compressed-air engine; gasoline, Diesel, or electric motor; or a line shaft. Field conveyors, as commonly installed, are operated through speed reducers from electric motors. Speed may be reduced through belts and pulleys, gears, rope drives, or chain and sprockets. Enclosed geared speed reducers are usually employed on field conveyors.

The power may be applied to the belt at the head pulley, at the tail pulley, or at any convenient intermediate point. Field conveyors are commonly driven from the discharge or head pulley and less often from the tail pulley, and where high belt tension is necessary the drive is usually independent of the conveyor terminal pulleys.

The power required to drive a belt conveyor depends upon the total weight moved, the frictional resistance to movement, and the height through which the pay load is elevated.

The total weight moved is the sum of the weights of the pay load, the belt itself, the moving parts of the supporting rolls, and all terminal, tension, and drive pulleys. The weight of the pay load depends upon its average cross-sectional area, its weight per cubic foot, and the length of the conveyor. The weight of the belt depends upon its width, length, the number of plies of duck, and the thickness of its rubber cover. The weight of the various pulleys depends on their size and the material of which they are built. The weight of the supporting rolls depends on the material of which they are constructed (steel tubing or cast iron) and their spacing along the conveyor.

The frictional resistance depends upon the type of bearings used in the several pulleys and rolls. This ranges from a simple shaft and pillow block through various types of roller or ball bearings to the latest type of improved antifriction construction. The frictional resistance depends also upon the type of lubricant used and the efficiency of application. Accurate calculation, based on a careful analysis of weight and friction coefficients, may be entirely erroneous in practical application if the lubrication is faulty.

The calculation of power required to elevate the pay load involves merely the weight lifted and the elevation. Sometimes calculation of this part of the power required may involve a percentage of the weight of the belt on the inclined portion of the conveyor, but the effect is so small that it is usually neglected.

If the conveyor operates on a descending rather than an ascending grade, the power developed by the load moving down-grade must be deducted from that required to move the load and overcome friction. If the power so generated is more than that required to move the load and overcome friction, some braking or hold-back device must be applied to control the belt speed.

In calculating the power required to move the belt it is customary to assume that the belt is uniformly loaded to its maximum average capacity. While it is true that in practical operation few belts are so loaded continuously the installed power must be sufficient to provide for such conditions.

From the preceding discussion it is evident that an accurate computation of power requirements can be made only when all the facts are known, and then only as they affect each installation. Conveyor-belt manufacturers use various formulas for computation of the required power. If a given set of operating conditions is assumed the power requirements calculated from the formulas may vary as much as 10 percent, probably owing to the variation in the values assigned to the average weight of the moving parts, the coefficient of friction, and the weight of the belt itself.

Calculation of power requirements by formulas must then be considered approximate rather than exact, unless all factors are accurately known. For this reason the author prefers to establish the minimum and maximum limits for variable factors, from which can be determined the minimum and maximum limits of the required power.

The range in weight of the moving parts of belt conveyors, calculated per foot of conveyor length and including 2 feet of belt, has been abstracted from manufacturers' catalogs and is shown in table 67.

TABLE 67.- Weight of the Moving parts of conveyor belts, including 2 feet of belt

Belt width, inches	Weight per foot of conveyor, pounds	
	Minimum	Maximum
12	12	15
14	15	17
16	17	20
18	19	24
20	21	28
24	25	35
30	32	47
36	40	63
42	49	76
48	59	91
54	69	107
60	81	125

The factors of frictional resistance, as determined empirically for conveyor belts, are as follows:

For plain greased bearings - - - - - - - - - - -	0.06
For average antifriction bearings - - - - - - - -	.04
For improved antifriction bearings - - - - - - -	.025

Table 68 shows the weight of the average maximum load per foot of conveyor, computed from the capacities given in table 66.

TABLE 68.- Weight of load per foot of conveyor, pounds

Belt width, inches	Weight of material per cubic foot, pounds				
	100	107.4	110	115	120
12	7.7	8.2	8.4	8.8	9.2
14	10.5	11.3	11.6	12.1	12.7
16	14.0	15.0	15.3	16.0	16.7
18	17.8	19.1	19.6	20.5	21.4
20	22.2	23.8	24.4	25.5	27.0
24	32.7	35.0	36.0	37.3	39.2
30	52.3	56.3	57.7	60.3	63.0
36	77.7	83.7	85.7	89.3	93.3
42	109.0	115.7	119.7	125.0	131.0
48	146.0	157.0	160.7	168.0	175.0
54	189.0	203.3	208.7	218.0	227.0
60	240.0	258.0	264.0	276.0	288.0

The power required depends directly upon the force or pull necessary to move the belt and its load. This pull may be divided into three components:

(1.) That necessary to move the empty belt horizontally;
(2.) that necessary to move the load horizontally; and
(3.) that necessary to move the load vertically.

For extreme accuracy a fourth component should be considered, namely, that necessary to raise the empty belt on the inclined portion of the conveyor, but this is small and is customarily omitted.

The horizontal pull required is computed by multiplying the total weight moved by the friction factor as follows:

$$hP = W \times L \times F,$$

in which

hP = horizontal pull, in pounds;
W = total weight per foot of conveyor, in pounds;
L = length of the conveyor, in feet;
F = factor of frictional resistance.

Then

$$W = w + m$$

in which

w = the weight of load per foot of conveyor (table 68), and
m = the weight of moving parts per foot of conveyor (table 67).

The vertical pull required by the load on 1 foot of the conveyor is found by multiplying its weight by the vertical distance through which it moves. Thus

$$vP = wH,$$

in which

vP = the vertical pull required to move the load carried on 1 foot of the conveyor, pounds; and
H = the vertical distance the load is elevated, feet.

The total pull required to operate the conveyor depends upon whether the conveyor is level, elevates the load, or lowers the load.

Total pull $P = hP$ for horizontal conveyors.

Total pull $P = hP + vP$ for elevating conveyors.

Total pull $P = hP - vP$ for lowering conveyors.

The power required may be found from the formula

$$HP = \frac{P \times S}{33{,}000},$$

in which

HP = theoretical horsepower;
P = total pull required, pounds;
S = speed of the conveyor, feet per minute.

Example 11

What horsepower would be required to operate a belt conveyor 36 inches wide and 500 feet long, using average antifriction bearings, running 300 feet per minute, and carrying sand and gravel weighing 110 pounds per cubic foot up an incline with a total elevation of 50 feet?

From table 67 the weight (m) of the moving parts may range from 40 to 63 pounds per foot. The weight is assumed as 50 pounds.

From table 68 the weight (w) of the load is 85.7 pounds per foot. Then

$$W = 85.7 + 50 = 135.7 \text{ pounds.}$$

Moreover,

$$\begin{array}{lllr} & & & \underline{\text{Pounds}} \\ hP = W \times L \times F & = 135.7 \times 500 \times 0.04 & = & 2,714 \\ vP = wH & = 85.7 \times 50 & = & \underline{4,285} \\ P = & & & 6,999 \end{array}$$

Then

$$HP = \frac{6,999 \times 300}{33,000} = 63.6$$

If the conveyor had been level the power required would be

$$\frac{2,714 \times 300}{33,000} = 24.7 \text{ HP}$$

If the conveyor lowered the gravel 50 feet the power required would be negative, or the power developed would be

$$\frac{(4,285 - 2,714) \times 300}{33,000} = 14.3 \text{ HP.}$$

These calculations do not include the power required to overcome friction in the driving mechanism. For enclosed-geared speed reducers and for each reduction by cut gears or belt and pulley 5 percent should be added for horizontal or elevating conveyors and subtracted for lowering conveyors. In addition, 10 percent should be added for surplus power to take care of temporary overload in determining the brake horsepower of the motor for the horizontal or elevating conveyor and subtracted for the lowering conveyor.

Belt Ply

The pull necessary to move the belt and its load horizontally, plus the pull necessary to elevate the load, constitute the "effective" pull or tension required to operate a belt conveyor. However, they do not equal the total tension that the belt must withstand. A conveyor belt may be visualized as reacting the same as any belt used to transmit power. If the belt is merely laid over the drive pulley without tightening, the pulley will revolve and slip beneath the belt without moving it. If the two ends of the belt are fastened together around another pulley and the belt is tightened, the belt will revolve with the drive pulley due to the friction between the belt and the face of the pulley. As the tension on the belt is increased the pulley friction increases and the belt's capacity to transmit power also increases. This tension, called "initial tension", is necessary before motion takes place in the belt and is independent of the tension necessary to move or elevate the load. However, increasing pulley friction by belt tension alone may set up undue stresses in the belt itself, and the tension must be limited by the strength of the belt which depends on the number of plies of duck it contains.

Pulley friction may be increased in two ways without increasing belt tension: (1) By increasing the arc of contact between drive pulley and belt and (2) by lagging the face of the drive pulley with rubber or some other material of a greater friction coefficient with the belt than the bare pulley face. Thus, an increase in pulley friction by either or both methods will permit application of more power without any increase in belt tension. This is an important factor in belt-conveyor design, since the construction of the belt and its ability to transmit power will depend upon the type of drive used.

Single, unlagged pulleys used without snubbers afford an arc of contact of only 180° and a minimum of friction. By using snubber pulleys the arc may be increased to 240°. Double-pulley or tandem drive affords an arc of contact of 360°, which may be increased to 430° or more by the use of snubbers. The friction in both types of drive may be increased by lagging the pulley faces.

The use of both tandem drives and snubber pulleys, however, flexes the belt in opposite directions, and the belt wraps around the second pulley with its load-carrying side in contact. This increases internal stress due to flexure and provides a chance for excessive abrasion from fine particles clinging to the belt and being ground against the pulley face. Abrasion is reduced if the pulleys are lagged.

Other types of drive have been developed for conveyor belts but those mentioned are in most common use.

The total tension to which a conveyor belt is subjected is then

$$T = P + S,$$

in which

T = total belt tension, pounds;
P = total belt pull or effective tension, pounds;
S = tightening or slack-side tension.

The value of S, or the slack-side tension in the belt, is produced by a counterweighted take-up device or by a screw take-up and is usually represented by a percentage of the effective tension, P; it depends upon the arc of contact with the drive pulleys and the coefficient of friction for bare or lagged pulleys.

With a screw take-up for applying tension there is no way of measuring accurately the tension applied, whereas with a counterweighted take-up the tension can tbe controlled by the weight.

Table 69 gives the average slack-side tension for counterweighted take-up drives in percentage of effective tension or belt pull.

TABLE 69.- Average slack-side tension in percentage of effective tension

Arc of contact, degrees	Counterweighted take-up	
	Bare pulleys, percent	Lagged pulleys, percent
180	100	52
210	70	40
240	54	32
360	26	13
420	20	9
480	16	6

From the values given in table 69 for slack-side tension, S, and the working stress of belt plies in table 61, the necessary number of plies and weight of duck can be computed by the following formula.

$$PD = \frac{T}{WX},$$

in which

PD = number of plies of duck;

T = total belt tension, in pounds;

W = width of belt, in inches;

X = ply-inch working stress of belt, in pounds.

Example 12

In example 11 the effective tension or belt pull was calculated to be 6,999 pounds. With application of a single, bare, unsnubbed head-pulley drive the total belt tension would be

$$T = P + S = 6{,}999 + (1.00 \times 6{,}999) = 13{,}998 \text{ pounds.}$$

Using 28-ounce duck with a maximum ply-inch stress of 25 pounds, the number of plies required would be

$$\frac{13{,}998}{36 \times 25} = 15.5$$

From table 58 the maximum number of plies permissible for proper troughing with a 36-inch belt was 9, hence some other type of drive will be required.

If a tandem drive is assumed with an arc of 420°, obtained by snubbing and lagged pulleys, the total tension will be

$$T = P + S = 6{,}999 + (0.09 \times 6{,}999) = 7{,}629 \text{ pounds.}$$

With 28-ounce duck at 25 pounds ply-inch stress the number of plies required would be

$$\frac{7{,}629}{36 \times 25} = 9.$$

PUMPS AND PIPE LINES

Centrifugal pumps are used at many plants to convey sand and gravel through pipe lines. The pump and its discharge line may constitute the sole method of transportation, as in a dredge pump delivering direct to the treatment plant. It may be supplementary to other means of transportation, as in trucks hauling from a dry pit and dumping to a sump from which the pump delivers the gravel to the plant. Another instance is conveyance from pit to sump by sluices supplemented by pumps from sump to plant. Conversely the pump and pipe line may comprise the primary means of transportation, supplemented by other types as in a dredge pump delivering to a dewatering bin on shore from which a conveyor belt takes the gravel to the plant.

Pump and pipe line transportation may be designed so that two or more pumps operate in series in a single pipe line. The dredge pump in this case is supplemented by one or more booster pumps in the discharge line. Under different conditions the dredge pump may discharge to sump or underwater storage and a second pump take material from there to the plant, thus cutting the haulage service into two or more independent units. A dragline, power scraper, or slackline cableway may be substituted for the last pump. Obviously, the centrifugal pump may be used in a number of ways.

The selection of pumps for transportation depends upon a number of factors such as: (1) The quantity of material to be transported, (2) the presence of enough water as a transporting medium, (3) the length of the haul, (4) the height to which the material must be elevated, (5) the physical character of the gravel, (6) the power required, and (7) the cost of installation.

The quantity of material to be transported obviously governs the size of pump and pipe line. The movement of quantities in excess of a certain limit - determined by local conditions - results in excessive expense for maintenance and power over that required for other haulage equipment, such as barges.

TABLE - 70. - Composition of sand, gravel, and water mixtures

(Sand and gravel assumed to weigh 110 pounds per cubic foot and contain 40 percent voids

Sand and gravel by volume, percent	100 cubic feet of mixture will contain -					1 cubic yard of sand[1/] and gravel will require water as follows-			Ratio of sand and gravel to water			
	Sand and gravel		Water						By volume		By weight	
	Cubic feet	Pounds	Cubic feet	Pounds	U. S. gallons	Cubic feet	Pounds	U. S. gallons	Sand and gravel	Water	Sand and gravel	Water
5	5	550	97.0	6,060	726	524	32,750	3,920	1	19.4	1	11.0
6	6	660	96.4	6,020	721	434	27,120	3,245	1	16.0	1	9.1
7	7	770	95.8	5,990	716	370	23,120	2,770	1	13.7	1	7.8
8	8	880	95.2	5,950	712	321	20,070	2,400	1	11.9	1	6.8
9	9	990	94.6	5,910	708	284	17,750	2,125	1	10.5	1	6.0
10	10	1,100	94.0	5,875	703	254	15,875	1,900	1	9.4	1	5.3
11	11	1,210	93.4	5,840	698	230	14,375	1,720	1	8.5	1	4.8
12	12	1,320	92.8	5,800	694	209	13,050	1,563	1	7.7	1	4.4
13	13	1,430	92.2	5,760	690	192	12,000	1,435	1	7.1	1	4.0
14	14	1,540	91.6	5,725	685	177	11,050	1,324	1	6.5	1	3.7
15	15	1,650	91.0	5,690	681	164	10,250	1,226	1	6.1	1	3.45
20	20	2,200	88.0	5,500	658	119	7,440	890	1	4.4	1	2.5
25	25	2,750	85.0	5,310	636	92	5,750	688	1	3.4	1	1.9
30	30	3,300	82.0	5,120	613	74	4,625	554	1	2.7	1	1.55
40	40	4,400	76.0	4,750	568	51	3,190	381	1	1.9	1	1.08

[1/] 1 ton of sand and gravel will require approximately two thirds the tabulated figures.

In handling sand and gravel through pipe lines the transporting medium is water. Under average operating conditions the sand and gravel conveyed constitute 10 to 15 percent by volume of the mixture pumped, the balance being water. Sand and gravel contain voids or unfilled spaces between particles. The percentage of voids varies with the size gradation of the gravel and sand. When the mixture is moved by pumping these voids are filled with water, hence the actual quantity of water present in a given volume is always greater than the complementary percentage of sand and gravel. If sand and gravel are assumed to average 110 pounds per cubic foot loose measure and to contain 40 percent voids the relative quantities of water and gravel for various mixtures is as given in table 70. After the percentage of solids is once determined the water required as a vehicle can be read from the table. Correction should of course be made for variation in weight and void content of the gravel. When gravel is dredged from a running stream there is assurance of enough vehicular water. When it is dredged from an artificial pond in which the subsurface water movement is retarded additional water may be necessary from an independent source. Again, pumps handling material from a sump may require more water than enters the sump with the gravel, although ordinarily the feed to the pump can be made more uniform, resulting in a heavier percentage of solids and requiring less water than in original pumping or sluicing.

The head against which the material must be pumped is affected by the length of the pipe line and the height to which the material must be elevated. The length and diameter of the pipe determine the friction head, and the elevation determines the static head.

The physical characteristics of the sand and gravel will affect the pumping operation in various ways, all of which, while evident, are difficult to evaluate accurately. Round particles create less internal pipe friction than flat or sharp-edged particles; coarse gravel creates more friction than sand and requires a higher velocity of flow; and increased percentages of solids, either coarse or fine, increase friction and require greater velocity. The direct effect of these characteristics must be determined by trial or from the past experience of the operator.

The power required is determined by the quantity pumped, the percentage of solids, the total pumping head, and the operating characteristics of the pump. Power is probably the factor of greatest economic importance in the selection of a pump and pipe lines as a means of transportation, as is illustrated in the following example:

Example 13

It is assumed an operator is pumping a mixture containing 12 percent solids against a total head of 25 feet at the rate of 100 tons of sand and gravel per hour and wishes to increase the head to 100 feet and put the material directly to the top of his washing plant.

If the sand and gravel weigh 110 pounds per cubic foot the required quantity (100 tons per hour) in cubic feet per minute will be

$$\frac{100 \times 2000}{60 \times 110} \text{ or } 30.3 \text{ cubic feet per minute.}$$

From table 70, 30.3 cubic feet of sand and gravel at 12 percent solids will require

$$12 : 92.8 :: 30.3 : X = 234 \text{ cubic feet of water.}$$

Assuming 40 percent voids and allowing for the water filling the voids the volume of mixture pumped per minute will be

$$30.3 - \frac{40 \times 30.3}{100} + 234 = 252 \text{ cubic feet.}$$

The weight of the mixture per cubic foot will be

	pounds
30.3 cubic feet of gravel at 110 pounds	3,333
234 cubic feet of water at 62.5 pounds	14,625
Total	17,958

Then

$$\frac{17,958}{252} = 71.3 \text{ pounds per cubic foot.}$$

The theoretical or water horsepower required can be computed from the formula

$$HP = \frac{Q\ W\ H}{33,000},$$

in which HP = the theoretical or water horsepower,

Q = the quantity of mixture handled, in cubic feet per minute;
W = the weight of the mixture per cubic foot;
H = the total head, in feet.

If there is no increase in the length of the discharge pipe required by the increase in total head substitution in the formula gives the following power requirements:

$$HP = \frac{252 \times 71.3 \times 25}{33000} = 13.6, \text{ to pump the mixture against a 25-foot head.}$$

$$HP = \frac{252 \times 71.3 \times 100}{33000} = 54.4, \text{ to pump the mixture against a 100-foot head.}$$

The motor rating in each case should be about double these figures. The operator will then be faced with the problem of changing from a 30-to a 110-HP motor. He must evaluate this against the economics involved in dewatering his pump discharge and using some other type of elevator to gain the added 75 feet of elevation. Possibly a belt conveyor or a bucket elevator would do the work with less than the 80 horsepower involved. On the other hand, he may be transporting water to the top of his plant for washing purposes.

It is interesting to note that if the gravel could be transported dry the power required would be only about 20 percent of that necessary to convey the mixture of gravel and water.

$$\frac{30.3 \times 110 \times 25}{33000} = 2.5 \text{ HP to pump gravel only against 25-foot head.}$$

$$\frac{30.3 \times 110 \times 100}{33000} = 10.1 \text{ HP to pump gravel only against 100-foot head.}$$

In other words, 80 percent of the power is used to transport the water.

Besides these factors, others may influence the operator in choosing this type of transportation equipment. Many gravel deposits contain objectionable quantities of sticky clay, which must be removed to meet specifications for washed aggregates. The churning action given the material in its passage through pump and pipe line and the solvent effect of the water provide a very effective preliminary washing. In some instances no further wash water may be required. In others, further rinsing may be necessary, but less additional water will be needed than if the gravel had been delivered dry to the plant.

Pumps and pipe lines share with sluices the advantage of being the only types of transportation which perform the dual function of washing and transportation.

Thorough wetting of the sand and gravel particles before their delivery to the treatment plant facilitates treatment in the plant. Thoroughly wet material accompanied by a relatively large body of water requires less time in screening. The same plant fed by dry material must provide some method of wetting either on the initial screen or before the material reaches it.

On the other hand, pump and pipe-line transportation probably requires more water to be handled than would be needed for washing only. Welch[5] estimates the average water requirement for washing in gravel plants as 1 gallon per minute per cubic yard per 10 hours of plant capacity but adds that more water is better. A study of a number of plants by the author reveals that the ratio of sand and gravel to water for washing ranges from 1 ton of sand and gravel to 1.2 to 4.1 tons of water.

5/ Welch, F. M., Design of Sand and Gravel Washing and Screening Plants: Rock Products, vol. 32, no. 15, Aug. 31, 1929, p. 52.

Applying Welch's average requirements to example 13 the water required would be 667 gallons per minute, and the ratio of sand and gravel to water by weight would be 1 to 1.7. Pumping with 12 percent of solids, the ratio of sand and gravel to water in the pipe discharge (from table 70) would be 1 to 4.4 by weight or 2-1/2 times the average quantity calculated. However, all the vehicular water cannot be used for washing, and additional water must be added for rinsing.

Pump and pipe-line transportation is advantageous for relatively short distances or average hydraulic heads. Procuring and delivering the gravel in one operation often eliminate intermediate storage and rehandling. The gravel is prewashed before it enters the washing plant, and a constant supply is provided to the treatment plant.

This method of transportation has the disadvantage of requiring a large proportion of additional power to transport the carrying water. It may be expensive due to the maintenance of floating pipe lines in river currents or in stormy weather.

Structural Limitations

Conveyance by pumps and pipe lines is limited to those gravel deposits which are comparatively free from oversize boulders. Ordinarily, with a deposit containing 10 percent of boulders larger than two thirds the diameter of the suction inlet of the pump this type of transportation could not be used.

No definite length of pipe discharge can be assigned for pipe-line transportation by a single pump or for one or more booster pumps in the line. The discharge length will depend upon the operating characteristics of the pumps used, the diameter of the pipe, and the character of the material pumped. However, certain limitations may be mentioned. Where booster pumps are necessary the material should enter the suction of the booster at some positive pressure. In other words, the booster should not be called upon to overcome any suction head. If the suction of the booster is put under a vacuum, leakage of air would probably result, causing pulsation in the flow. The amount of pressure required is only that necessary to insure against any vacuum in the booster suction at any time and is estimated by various engineers as ranging from 4.3 to 6.0 pounds per square inch, corresponding to 10 to 14 feet of head.

It is necessary also that the booster pump be so located that the discharge head or pressure is well within the bursting strength of pipe and pump. Sand and gravel traveling in long, straight, fairly level pipe lines tend to separate and drag along the bottom of the pipe. This may cause the line to fill up gradually and plug. The booster pump should therefore be placed as near the center of the discharge line as possible so that it may reagitate the material and prevent plugging. With a booster pump, if for any reason the dredge pump is stopped and there is no positive head on the suction of the booster, some means of relief must be provided or the line beyond the booster will also plug. Provision is commonly made for this by interlocking

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the drives of both pumps or by inserting a bypass valve between the booster and the dredge pump. This valve is placed in a submerged depression in the discharge line with its emergency opening under water. Ordinarily the pressure in the pipe line holds this valve closed. When the pressure is removed by stoppage of the dredge pump or a plug in the line the valve opens and allows the booster pump to pull clear water into its suction and thus clear the line beyond. Such valves do not help if the stoppage occurs in the booster discharge line, hence interlocking drives are needed.

It has been shown that the power required to operate a pump and pipe-line transportation unit varies directly with: (1) The rate of discharge - governed by the diameter of the pipe and the velocity of flow, (2) the weight per cubic foot of the material pumped - governed by the percentage of solids, and (3) the total head against which the pump must operate - governed by the suction head, the friction head, and the static head. The head against which a pump can deliver depends upon the peripheral speed of its impeller, which in turn depends upon the diameter and revolutions per minute of the impeller or, in other words, the characteristics of the pump.

The total head against which a pipe-line conveyor operates may be constant or may vary from day to day. To illustrate, a pump obtaining material from a sump, supplied by other equipment, and transporting it to the treatment plant will operate under constant head. On the other hand, delivery from a constantly shifting dredge may or may not be subject to a corresponding change in head, depending on whether shifting the position of the dredge involves lengthening or shortening or no change in the discharge line. Consequently, a pump and pipe-line system subject to frequent change of head should be provided with means for varying the peripheral speed of the pump impeller. This is usually accomplished by variable-speed reduction between motor and pump. As an illustration of the seriousness of the effect of change of head Hawgood[6] found that a pump handling sand and mud and operating at highest efficiency through a 4,000-foot discharge line required 10 HP per 100 feet of discharge line. When the discharge line was cut to 250 feet but the revolutions per minute of the pump were left as before, the power required rose to 108 HP per 100 feet of pipe. In other words, the efficiency of the pump varies inversely with the ratio of the peripheral speed of the impeller to the velocity equivalent of the total head, and highest efficiency results when the impeller speed approaches the velocity equivalent of the total head.

For a detailed discussion of the effect of design on pump efficiency the reader is referred to an article entitled "Economic Design of Hydraulic Pipe-Line Dredge," by John F. Cushman, published in Proceedings of the American Society of Civil Engineers, vol. 56, November 1930.

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[6] Hawgood, H., The Hydraulic Transmission of Dredged Material at San Pedro Harbor, Calif.: Eng. News, vol. 62, no. 10, Sept. 2, 1909, pp. 244-6.

Auxiliary Equipment

Pump and pipe-line systems require a source of power. This source may be a steam engine or turbine or gasoline, Diesel, semi-Diesel, or electric motor. Combinations of Diesel, semi-Diesel or steam turbines with electric generators and motors may also be used.

The necessary priming apparatus is required, the characteristics of which depend upon the design of the individual pump.

While the pipe line is auxiliary to the pump, it is part of the system. Pipe lines, however, may be installed in various ways. They may be laid on the ground surface, floated on a water surface, or submerged and laid on a lake or river bed. In any case they must be flexible to provide facility in movement from place to place and to prevent breakage which is apt to occur from surges in the material pumped. On the other hand, adequate but flexible anchorage should be provided to prevent creeping of submerged or land lines.

Flexibility in the line is provided by rubber sleeves or balljoint connections between pipe sections at regular intervals. Excessive movement is prevented by fastening to "dead men" on the surface or anchors in the river bed. Floating lines are snubbed to anchored buoys.

Rubber-hose joints offer greater resistance or internal friction than ball joints but are less expensive in first cost. They are more liable to failure than ball joints but are usually replaced in less time when failure occurs. They are seldom used in modern practice on pipe larger than 15 inches in diameter.

Flexible joints on floating pipe lines should be placed between pontoon supports and not on the pontoon itself.

Pontoons are placed at various intervals in the line, depending on local conditions. Pipe is usually furnished in 20- to 40-foot lengths with flanged joints. Pontoons may be 30 to 60 feet apart but usually at not more than 40-foot centers.

The discharge pipe may be carried either parallel to the long axis of the pontoon or transverse. Pontoons of wooden or steel cylinders are usually built with two cylinders connected by a wooden or steel frame supporting the pipe cradle.

There is no uniformity in pontoon design, but from observed installations the ratio of the weight of the water which the pontoon is capable of displacing to the total weight supported ranges from 1.4 to 1 to 3.0 to 1. The pontoon must support its own weight, plus that of the pipe, plus that of the gravel and water in the pipe, hence if the weight of these three items is 1,000 pounds the pontoon should displace from 1,400 to 3,000 pounds of water when completely submerged.

Reference to the variation in weight between standard and spiral pipe, as given in table 72, will indicate that a pontoon suitable for spiral pipe may not be large enough for standard pipe.

Discharge lines offer the least frictional resistance when straight and when the ends at the joints are brought together without leaving gaps within the coupling or flange. All elbows and bends should be on as large a radius as possible.

The frictional resistance of various pipe-line accessories, expressed in equivalent feet of straight pipe, is given in table 71. For pipe friction and capacity and for head developed by dredge pumps the reader is referred to tables 32 to 38 in part 3 of this series of papers. Table 72 gives the principal characteristics of pipe commonly used in dredging service.

TABLE 71. - Friction equivalent (in feet of straight pipe) of loss in fittings

Fitting	Diameter of pipe, inches											
	4	5	6	8	10	12	14	15	16	18	20	24
Elbow 90° long	9.5	11	12	16	20	24	28	30	32	36	40	48
Elbow 90° short	14	16	18	24	30	36	42	45	48	54	60	72
Elbow 45°	7.2	8.4	9	12	15	18	21	23	24	27	30	36
Elbow 22½°	4.8	5.5	6	8	10	12	14	15	16	18	20	24
Gate valve	9.6	11	12	16	20	24	28	30	32	36	40	48
Flap valve	19	22	24	32	40	48	56	60	64	72	80	96
Pontoon connection	12	14	15	20	25	30	35	38	40	45	50	60

TABLE 72. - Characteristics of dredge pipe

Internal diameter, inches	Standard lap welded pipe			Spiral pipe			
	Thickness, inch	Weight per foot, pounds	Bursting strength, lb./sq.in.	Thickness, U.S. gage	Weight per foot, pounds	Theoretical safe working head, feet	Bursting strength, lbs./sq.in.
6	0.28	19.0	[1]1,000	16	5.3	400	1,250
8	.322	28.5	[1]1,000	16	7.1	346	935
10	.366	40.5	[1]1,000	16	8.8	276	750
12	.375	49.5	3,150	16	10.6	230	625
15	.375	65.0	2,500	14	17.0	230	625
16	.375	69.4	2,344	14	18.1	216	585
18	.375	77.8	2,082	14	19.9	192	525
20	.5	116.2	2,500	14	22.1	172	470
24	.5	138.0	2,082	12	36.5	201	540
26	.5	150.0	1,922	12	39.5	184	505
30	.5	172.0	1,666	10	56.8	208	560

[1] Routine factory test.

A choice between the use of spiral or standard pipe will depend largely upon the experience of the operator. Standard pipe of equal diameter, as will be noted, is much heavier; it will also withstand much higher heads or pressures, but spiral pipe is strong enough for average dredging requirements.

Standard pipe, on the other hand, will withstand internal abrasion and wear longer than spiral pipe. Where discharge lines are carried over land or can be held in a fixed position it is sometimes possible to revolve them one third or one half and thus prolong their life against abrasion from sand and gravel dragging along the bottom.

Where it is necessary to submerge the pipe line to the river or lake bottom as a protection against swift currents, storms, or surface traffic, the pontoons may be filled with water and sunk. When it is necessary to move the line, the water is blown out with compressed air and the line floated.

Capacity

The capacity of a pump and pipe-line system depends upon the internal diameter of the pipe, the velocity of flow, and the percentage of solids in the mixture.

The diameter of the discharge line is ordinarily fixed by the discharge diameter of the pump.

The velocity of flow in pumping sand and gravel will seldom be less than 10 feet per second or more than 18 feet. The usual velocity averages 12 feet per second.

The amount of solids in the mixture is largely a function of the skill of the operator and the character of the gravel. It ranges from 5 to 20 percent by volume and usually averages 10 to 15 percent.

Table 73 gives the capacity of pump and pipe-line systems for the average range of velocities and percentages of solids. The figures in the table are based on sand and gravel weighing 100 pounds per cubic foot, loose measure. The capacity corresponding to heavier material will vary in direct proportion; hence, for sand and gravel weighing 110 pounds per cubic foot the figures should be increased 10 percent.

TABLE 73. - Capacity of pump and pipe-line systems, tons per hour[1]

Pipe diameter, inches	Solids, percent	Velocity of flow, feet per second								
		10	11	12	13	14	15	16	17	18
6	5	17.7	19.5	21.2	23.0	24.7	26.5	28.4	30.0	31.8
	10	35.4	39.0	42.3	46.0	49.5	58.2	56.7	60.0	63.6
	15	53.1	58.5	63.5	69.0	74.2	79.7	85.0	90.0	95.4
	20	70.8	78.0	84.7	92.0	99.0	106	113	120	127
8	5	31.3	34.5	37.7	40.8	43.8	46.9	50.0	53.3	56.4
	10	62.7	69.0	75.4	81.6	87.6	93.9	100	107	113
	15	94.1	104	113	122	131	141	150	160	169
	20	125	138	151	163	175	188	200	213	225
10	5	49.0	54.0	58.8	63.7	68.7	73.6	78.4	83.4	88.3
	10	98.0	108	118	127	137	147	157	167	177
	15	147	162	176	191	206	221	235	250	265
	20	196	216	235	255	275	295	314	333	353
12	5	70.7	77.7	84.8	91.8	98.8	106	113	120	127
	10	141	155	170	184	198	212	226	240	254
	15	212	233	254	276	296	318	339	360	382
	20	283	311	339	367	395	424	452	480	509
14	5	96.3	106	115	125	135	144	154	163	173
	10	193	212	231	250	269	288	308	327	346
	15	289	318	346	375	404	433	462	490	520
	20	385	424	462	500	538	577	616	654	693
16	5	126	138	151	163	176	188	201	214	226
	10	251	277	302	327	352	377	402	427	453
	15	377	415	453	490	528	565	603	641	679
	20	503	553	604	654	704	754	804	855	905
18	5	159	175	191	207	223	239	255	271	286
	10	318	350	382	414	446	478	510	541	573
	15	477	525	573	621	669	716	764	812	860
	20	637	700	764	828	892	956	1,018	1,082	1,145
20	5	197	216	236	256	275	295	315	334	354
	10	394	433	472	512	551	590	630	669	708
	15	590	650	708	768	826	885	945	1,000	1,060
	20	787	866	945	1,023	1,100	1,180	1,260	1,338	1,415
24	5	283	311	339	368	396	424	452	481	509
	10	566	622	679	735	782	848	904	962	1,018
	15	849	933	1,017	1,100	1,187	1,272	1,356	1,442	1,527
	20	1,130	1,244	1,357	1,470	1,583	1,696	1,810	1,923	2,036
30	5	330	363	397	430	463	496	528	562	594
	10	660	727	794	860	925	992	1,057	1,124	1,188
	15	990	1,090	1,190	1,290	1,388	1,487	1,585	1,686	1,783
	20	1,320	1,453	1,587	1,720	1,850	1,983	2,113	2,248	2,380

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[1] In this table sand and gravel is assumed to weigh 100 pounds per cubic foot. For material weighing 110 pounds per cubic foot increase figures 10 percent, etc.

To illustrate the method of calculating the equipment necessary for a gravel pumping plant and pipe-line transportation an assumed set of conditions is selected and the equipment calculated in example 14.

Example 14

It is assumed that the gravel deposit contains considerable clay requiring careful washing and that it is mined by hydraulicking and the gravel sluiced to a sump in the pit. The distance from sump to washing plant is 4,000 feet, and the top of the plant is 40 feet above the water level in the sump. The desired rate of production is 3,000 tons per 10-hour day. It is further assumed that the gravel particles are water-worn to rounded edges and that there are no troublesome boulders which cannot be eliminated by the hydraulic mining method. Because of the clay it is decided to pump the gravel from sump to washing plant.

As an initial calculation, as discussed in part 3 of this report under hydraulic dredges, the capacity of a dredge pump in cubic yards per hour when pumping 10 percent solids at a velocity of 12 feet per second is seven eighths the square of the diameter of the pump in inches. The desired capacity in this example is 3,000 tons, or 2,000 cubic yards per 10 hours, or 200 cubic yards per hour; then

$$200 = \frac{7}{8} d^2 ,$$

$$d = 15.1.$$

Therefore a 16-inch pump is indicated.

The total head against which the unit must operate is as follows:

	Feet
Static head	40
Friction head (4.7 to 6.6 feet per 100 feet of 16-inch pipe at 12 feet per second[7]) at average 5.6 x 4,000	224
Suction head assumed at	15
Total hydraulic head	279

Since 150 feet is the customary limit for a standard pump it is evident that a booster pump is necessary in the line. If the pipe line is constructed so that the grade is uniform from sump to discharge and the booster pump is in the center of the line, the head on each pump will be as follows:

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[7] Table 32, part 3 of this series, chapter on "Hydraulic Dredges."

Sump pump:	Feet
Suction head	15
Friction head at 5.6 x 2000 ft.	112
Static head	20
Positive head delivery to booster	12
Total	159

Booster pump:	
Friction head at 5.6 x 2000 ft.	112
Static head	20
	132
Less positive head from sump pump	12
Total head	120

To equalize the duty on both pumps the booster is placed 1,700 feet from the sump and 2,300 feet from the plant. The head on each pump is then as follows

Sump pump:	Feet
Suction head	15
Friction head 5.6 x 17	95.2
Static head	17.0
Pressure to booster	12.0
Total	139.2

Booster pump:	
Friction head (5.6 x 23)	128.8
Static head	23.0
	151.8
Less positive head from sump pump	12.0
Total head	139.8

Assuming that the pumps are identical in design and each has a 48-inch impeller, then each should operate at 414 r.p.m. to develop 140 feet of head (table 38, part 3).

The power required for each pump can be calculated from the formula

$$HP = \frac{QWH}{33000}$$

in which
Q = quantity of material, in cubic feet per minute;
W = the weight of the material per cubic foot;
H = the total head, in feet.

From table 70 a cubic yard of sand and gravel with 10 percent solids will require 254 cubic feet of water, making a total of 254+27 = 281 cubic feet of mixture per cubic yard of gravel. Since the pumps must handle 200 cubic yards of gravel per hour the quantity of material handled per minute will be

$$\frac{200 \times 281}{60} = 937 \text{ cubic feet.}$$

From table 34, part 3, the specific gravity of a mixture containing 10 percent solids is 1.12, therefore each cubic foot will weigh

$$1.12 \times 62.5 = 70 \text{ pounds.}$$

Substituting in the formula,

$$HP = \frac{937 \times 70 \times 140}{33,000} = 278.2 \text{ (water horsepower).}$$

The motor rating or brake horsepower required on each pump would then be approximately 600 HP.

It will be noted from table 73 that a 16-inch pipe line handling 10 percent solids at a velocity of 12 feet per second will deliver 302 tons per hour, or that required in the example.

In computing the total head for example 14, no allowance has been made for the additional friction due to valves, bends, and flexible couplings in the pipe line. Accurate calculations would require that they be included in computing total head.

TOWBOATS AND BARGES

The previous discussion of haulage units has been confined to their use in intraplant haulage, that is, transfer of material from excavator to treatment plant. Except for motor trucks the types of haulage so far mentioned are not adapted to delivery beyond the plant limits. While towboats and barges are commonly used for intraplant service, they are more often used to move material in interplant haulage, as between the treatment plant on the dredge and the storage yard or other plant on shore or direct to consumer.

Barges moved by towboats or tugs are the most popular type of conveyance serving sand and gravel dredges, and whether their use is confined to intraplant or extended to interplant haulage will depend on whether the dredge carries partial or complete treatment machinery. Dredges of the clamshell and dipper type do not ordinarily have washing and screening equipment aboard, and the barges serving them are restricted to intraplant haul. As previously noted, however, the hydraulic dredge may provide its own conveying system through pipe lines. The ladder dredge, on the other hand, is almost invariably equipped with washing and screening machinery, and the barges serving it are usually employed for interplant haulage. However, the wide geographic range possible with any type of dredge necessitates equipment capable of covering distances greatly in excess of those ordinarily covered by the previously discussed methods. Hence, most barge designs are such that they may be used for either intraplant or interplant haulage. The exceptions are where a dredge is used in an artificial pond which has been so enlarged that the distance is too great for direct pump and pipe-line delivery.

Barges may be the sole vehicle, they may be supplemented by other types of equipment, or they may themselves be the supplementary equipment. For example, the barge may be loaded directly by the dredge and towed to the treatment plant. If the treatment plant is on the shore line the dock equipment used to unload the barge may deliver directly to the plant. If the plant is some distance away, some supplementary form of conveyance is necessary. This may consist either of cars hauled by locomotive, autotrucks, an engine plane operated by a hoist, or a conveyor-belt system. Barges may also be unloaded by crane or other means to underwater storage, from which a hydraulic dredge, power scraper, or slackline cableway excavator may complete the transfer to the treatment plant.

In other instances a hydraulic dredge may excavate and deliver material directly to a centrally located, underwater storage from which barges may be loaded by various means for further movement. Barges are also used frequently for haulage between shore storage yards, between shore plant and scattered storage yards, or from either plant or storage to consumers delivery point.

As most used, sand and gravel barges have no means of generating power either for self-propulsion or for unloading their cargo. Recent development, however, has produced both self-propelled and self-unloading barges, but these developments are as yet confined to localities with specific need for such equipment.

The self-unloading type may be a simple rectangular barge with a crane wielding a clamshell bucket mounted on one end. The barge is loaded by the dredge, towed to its destination by towboat or tug, and left to unload its cargo while the tug proceeds on other business. Another type of self-unloading barge is equipped with conveyor belts feeding a boom conveyor which can be swung over the side to discharge to shore storage or other point as desired.

Self-propelling barges ordinarily are specially designed equipment for specific use or are converted lake or ocean steamers. They may or may not be capable of unloading their own cargo.

Barges without power are more or less standardized equipment. By this it is meant that manufacturers of this type of equipment have standardized on sizes and details of design to the extent that certain sizes and designs are built and carried in stock for quick delivery. Barges equipped with power are built or designed to meet specific conditions and are not standardized.

Structural Limitations

Sand and gravel barges for river or wet-pit service are usually rectangular in shape with vertical sides, and both ends are raked to provide minimum resistance to towing. Either wood or steel is used in their construction, the present tendency being toward steel, as it provides greater strength, less weight, and greater capacity per unit of weight.

Barges used in lake or ocean service are frequently of the same design but may have round bottoms with blunt-tapered ends.

Sand and gravel barges for pit, river, lake, or ocean service are constructed in three general designs, depending upon the position of the load with respect to the deck line. Hopper-type barges (fig. 24) carry the entire cargo below the deck. This type is used less extensively for gravel transfer. It has the advantage of providing a lower center of gravity when loaded and hence is more seaworthy. It has a disadvantage, however, in that the water entering the hopper with the gravel or from rain must be pumped out. Hopper-type barges are somewhat cheaper but are structurally weaker.

What may be termed a "semi-hopper-type barge" is designed so that the load is carried partly below and partly above the deckline. It is subject to the same objection as the full hopper barge in that water entering the hopper must be pumped out. It is less stable in stormy weather but somewhat stronger structurally.

The flush-deck type (fig. 25) of construction is probably used most for gravel haulage. On it the whole cargo is carried in a cargo box built on the barge deck. In the sides of the cargo box limber holes are provided through which rain water and water accompanying the gravel can drain off automatically, obviating the necessity for pumping. The flush-deck barge carries its entire cargo above the water line and is therefore the least seaworthy of the three types. However, it can be demonstrated that a properly designed barge with a hold free of water cannot be capsized with a reasonable load, although in a severe storm it may lose part of its cargo due to waves. If water is allowed to accumulate in the hold this type of barge is not seaworthy, but the accumulation of water in the hold is a matter of maintenance and should be prevented in any type of barge.

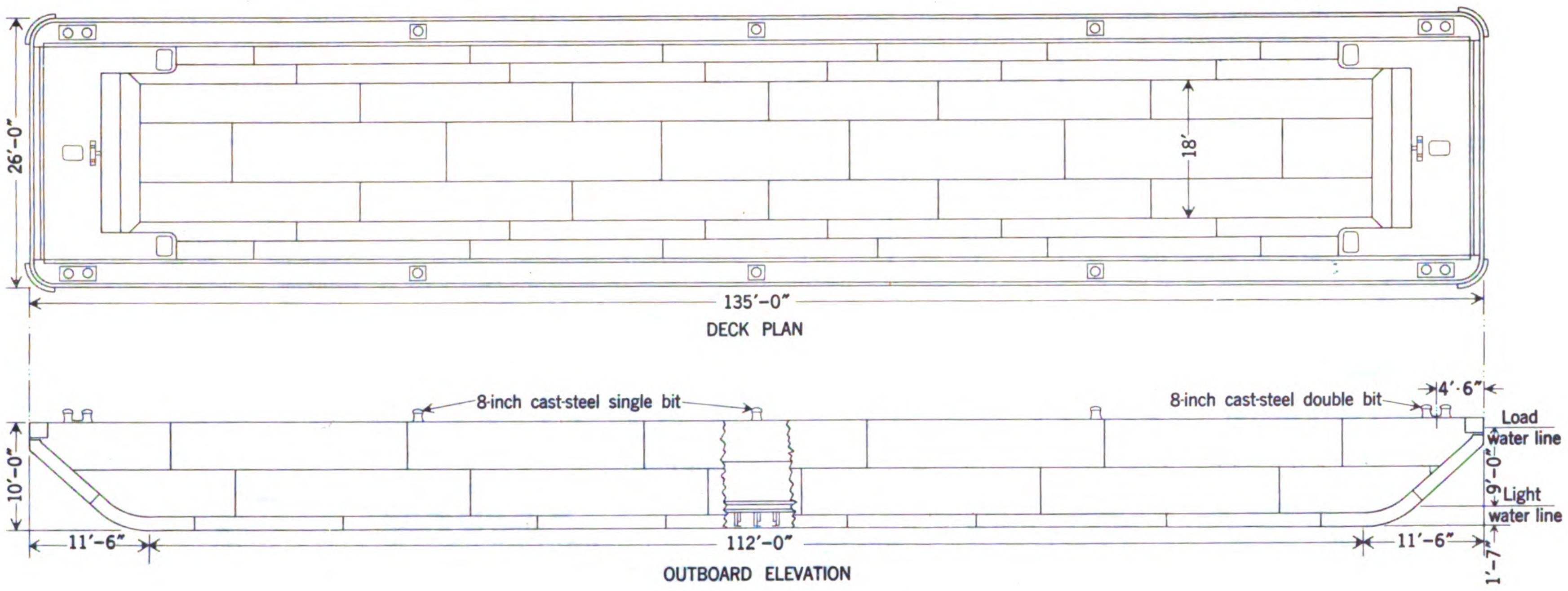
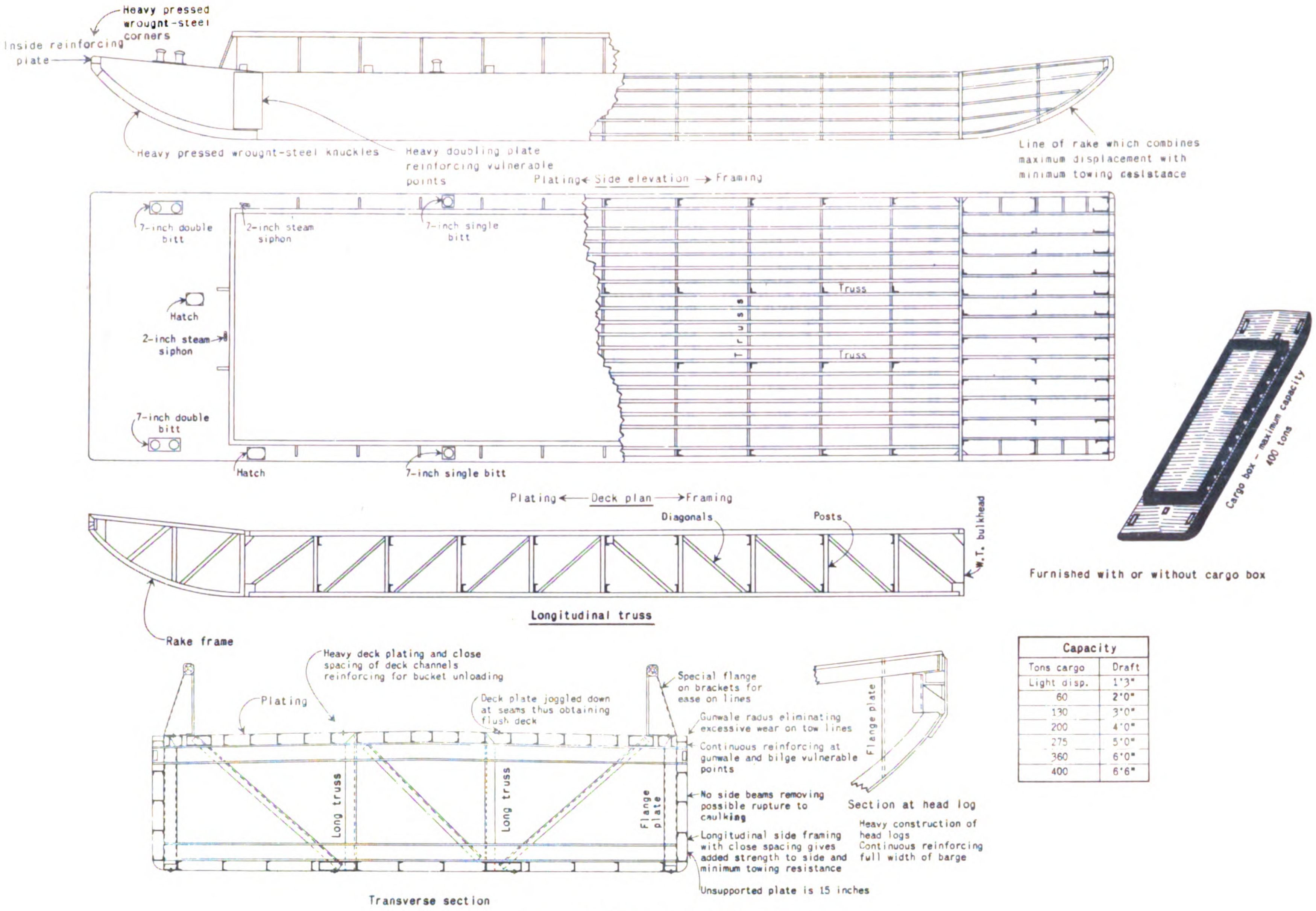


Figure 24.—Plan and elevation of hopper-type barge.


Heavy pressed wrought-steel corners
Inside reinforcing plate
Heavy pressed wrought-steel knuckles
Heavy doubling plate reinforcing vulnerable points
Plating ← Side elevation → Framing
Line of rake which combines maximum displacement with minimum towing resistance
7-inch double bitt
2-inch steam siphon
7-inch single bitt
Hatch
2-inch steam siphon
7-inch double bitt
Hatch
7-inch single bitt
Truss
Truss
Truss
Plating ← Deck plan → Framing
Diagonals
Posts
W.T. bulkhead
Longitudinal truss
Rake frame
Cargo box – maximum capacity 400 tons
Furnished with or without cargo box
Heavy deck plating and close spacing of deck channels reinforcing for bucket unloading
Plating
Deck plate joggled down at seams thus obtaining flush deck
Special flange on brackets for ease on lines
Gunwale radus eliminating excessive wear on tow lines
Continuous reinforcing at gunwale and bilge vulnerable points
No side beams removing possible rupture to caulking
Longitudinal side framing with close spacing gives added strength to side and minimum towing resistance
Unsupported plate is 15 inches
Long truss
Long truss
Flange plate
Transverse section
Flange plate
Section at head log
Heavy construction of head logs
Continuous reinforcing full width of barge

Capacity	
Tons cargo	Draft
Light disp.	1'3"
60	2'0"
130	3'0"
200	4'0"
275	5'0"
360	6'0"
400	6'6"

Figure 25.- Details of flush-deck type of barge.

For river or pit service the flush-deck type of barge is designed for only 6 inches of freeboard when fully loaded, although hopper-type barges require more. For lake or ocean duty more out-of-water height is desirable to withstand waves and storms.

Generally, the size of the barge is increased as the length of the haul increases, because the first cost is less per unit of capacity and large tows are handled easier in large units. Thus, for long hauls handling river freight, barges frequently are 2,000 or 2,500 tons in capacity. For the usual short or gravel-plant haul, however, barges seldom exceed 800 tons in capacity and may be as small as 100 tons. The smaller sizes are usually of wood, but these seldom exceed 100 feet in length or 400 tons in capacity, while many operators find steel the more economical structural material for capacities of 300 tons or more. As an example of the relation of barge size, capacity, and depth of draft, one manufacturer lists the following stock sizes in steel barges.

Length, feet	Beam, feet	Depth, feet	Capacity load, tons	Draft at capacity load, feet
100	26	6.5	365	6.0
130	30	7.5	635	7.0

It is not within the province of this paper to discuss manufacturing details of barge construction.

Auxiliary Equipment

The ordinary nonpower gravel barge requires some means of locomotion, and this is provided by towboats or tugs. Towboats of the side or stern paddle-wheel type (side paddles are now seldom used) are used in river service or wherever shallow drafts are necessary. They have rectangular hulls, raked bow and stern, and towing drafts ranging from 2 to 4 feet. Their towing speed depends upon the size of the tow and whether the loads must be transported against or with the river current. While a towboat may be capable of developing a speed of 12 to 15 miles or more per hour by itself in still water, this may be cut to 2 to 8 miles by the size of the tow or the river current. In fact, a towboat may be able to handle several barges without loads upstream, whereas one loaded barge would tax its power under the same conditions.

Side or stern paddle-wheel towboats are suited only to river or pit haulage.

For lake or ocean service, where shallow draft is unnecessary, tugs with round bottoms and screw propellers are usually employed to provide motive power.

Steam is probably the most popular type of power used, although gasoline-driven internal-combustion engines are common in the smaller sizes of towboats and tugs. Diesel-powered boats and tugs are a more recent development, and their use is increasing as old equipment is replaced.

The design and construction of towboats and tugs is a matter of special marine-engineering study, and this type of construction is not standardized. Usually each unit is built to specification for a particular service, although many boats, built for one service, have been used for others either without change or after remodeling to suit the new conditions. Their design requires such special knowledge that the sand and gravel operator will do best to leave this to reputable builders, contenting himself with stipulating the details of the service for which he desires the unit.

Self-propelled sand and gravel barges are really marine equipment specially designed for the sand and gravel trade, and their design and construction will depend entirely upon the service in which they are to be employed.

Self-unloading barges are equipped with various types of unloading machinery. The simple barge with crane and clamshell bucket mounted on one end has been mentioned. Conveyor belts are commonly used to deliver the cargo to shore or other floating equipment. To adapt the conveyor belt to this use the belt itself is usually mounted on a structural-steel boom capable of radial movement over the side of the barge. The boom belt may be loaded by power scrapers operating in the hold of the barge and delivering to a hopper over the inboard end of the conveyor, or it may be loaded by a belt conveyor taking its load from chutes in the bottom of the cargo hoppers. Other types of self-unloading barges omit the radial-boom conveyor and simply extend the hopper conveyor over the bow of the barge. This type of barge is moored with its bow to the dock, and the conveyor usually delivers to a hopper over another conveyor on the dock. In such barges the flush-deck cargo box is replaced by hoppers supported by steel framework above the deck on which the conveyor operates. The barge may generate its own power to operate the belt or may depend upon power lines on the dock to supply current to the conveyor motors. The Ward Sand & Gravel Co.[8] for wet-pit operation uses a barge (fig. 26) of this design in which the customary hull is replaced by 8 to 12 cylindrical steel pontoons 10-1/2 feet in diameter by 40 feet long. Such construction is of course adapted only to the quiet water encountered in pit operation.

Capacity

The capacity of a towboat and barge system of haulage depends upon a number of factors so variable that tables are impossible. The factors may be summarized briefly as the length of haul, the loaded and empty speed, the time required to load and unload barges, and the idle time due to waiting for towboats. The length of haul will of course vary with the position of the dredge with respect to the unloading point. For pit operations this will probably remain fairly constant but will increase gradually as the dredge advances away from the plant. For river operations the length of haul will be constant while the dredge operates over one bar. This may require a few days or the whole operating season depending on the size of the deposit. Where several bars containing materials of different physical characteristics

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[8] Ward, F. L., Methods and Costs of Mining and preparing Sand and Gravel at the plant of the Ward Sand & Gravel Co., Oxford, Mich.: Inf. Circ. 6580, Bureau of Mines, 1932, 16 pp.

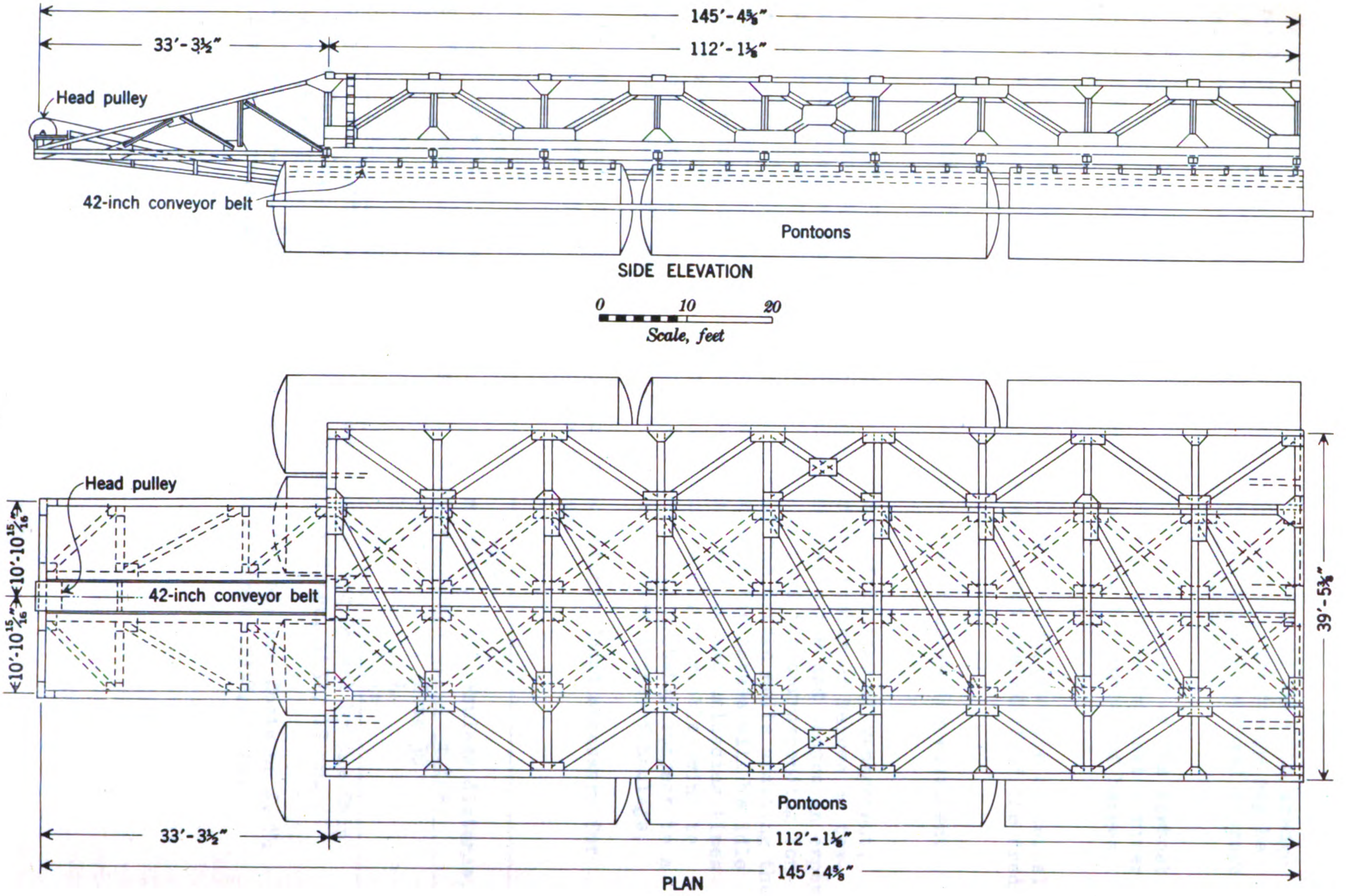

Figure 26.–Self-unloading barge, Ward Sand & Gravel Co.

are to be operated with one dredge it may be necessary to move the dredge frequently and thus vary the haulage distance. Again, the dredge may be stationary, but it may be necessary to deliver material to several storage or delivery points at unequal distances from the dredge.

The loaded and empty speeds will depend upon the power of the towboat and whether the barges are moved in quiet water or with or against a river or tidal current. The speeds will depend also upon the number of barges in the tow.

The time required to load the barge will depend upon the type of deposit being excavated, the type and size of the dredge, and the skill of the dredge runner.

The unloading time will depend upon the type of unloading equipment used.

The idle time will depend largely upon the efficiency of management, provided adequate equipment is available. If the number of barges or towboats is insufficient the time required to deliver a tow and return an empty may be more than that required by the dredge to load the corresponding tonnage. In that case barges must remain idle at the dredge while awaiting the towboat. If the unloading equipment is inadequate the barges will be idle at that end waiting to be unloaded. Since the loading and unloading times are themselves variable and the traveling time is also variable owing to different haulage distances, towboats and barges are probably subject to a greater percentage of unavoidable idle time than other types of haulage.

As examples of the diversity in the requirements found necessary for barge haul, three typical installations follow.

Number of barges	Barge capacity, tons	Dredge no. and type	Dredge capacity, tons per hour	Haulage distance, miles
20 1/	650	2 ladder	900	2 to 30
3 2/	3,000	1 pump	1600	½
48 3/	600	1 ladder	500	35

1/ Duffy, J. H., Method and Cost of Dredging Sand and Gravel by the Ohio River Sand Co.: Inf. Circ. 6421, Bureau of Mines, 1931, 17 pp.

2/ See footnote 8.

3/ Williamson, G. H., Sand and Gravel Dredging Methods and Costs of J. K. Davison & Bro.: Inf. Circ. 6582, Bureau of Mines, 1932, 10 pp.

AERIAL TRAMWAY

The aerial-tramway system of haulage, also known as a "ropeway", "wire-rope conveyor", or "cableway",[9] is a method of transporting material a few hundred feet or many miles by carrier buckets suspended in the air from moving or standing wire ropes which in turn are supported by wooden or steel towers.

There are several distinct types of aerial trams which have been developed by differences in local haulage conditions. The following are the principal types now in use:

1. Single-rope, fixed-clip tramway.
2. Monocable, or single-rope, saddle-clip tramway.
3. Bicable or double-rope tramway.
4. Jig-back or two-bucket, two-track, reversing tramway.
5. To-and-fro or single-bucket, single-track, reversing tramway.
6. Hoisting and transporting cableway.

In the first two types the buckets are attached to and suspended from endless moving ropes passing over sheaves on the supporting towers.

In the other four types the buckets are suspended from, and move over, standing track cables supported by towers and are attached to a traction rope which provides motion.

The single-rope, fixed-clip tramway consists of an endless moving rope to which the buckets are fixed at regular intervals. It is designed to run at low speeds carrying light loads. As the buckets are always attached to the running rope they must be loaded and unloaded while in motion.

The monocable, or single-rope, saddle-clip tramway consists of an endless moving rope to which the buckets are secured by friction grips built into the design of the saddle clip. It is designed for high speeds (up to 500 feet per minute) and heavy loads (up to 250 tons per hour). The bucket carriage is supplied with sheaves permitting movement over stationary rails and a V-shaped saddle clamp which grips the running cable securely. The buckets are detached automatically for loading and unloading and reattached automatically for despatch.

The bicable or double-rope tramway consists of an endless moving rope to which the buckets are manually or automatically attached and detached by friction clamps of various designs. The buckets are attached to carriages hung from sheaves which travel over standing track cables supported by towers. They are designed for high speeds and large individual bucket capacities. Loaded buckets are attached to the traction rope at specified intervals and are carried over one track cable to their destination, where they are usually detached for dumping and then refastened to the traction rope for return to the loading point over a second parallel track cable. Under favorable conditions it is unnecessary to detach the bucket for dumping. The buckets are attached to the running rope manually or automatically.

9/ A "cableway" strictly speaking, is a special form of tramway and should not be loosely classed as an aerial tram.

Under favorable conditions the buckets may be fixed to the traction rope and remain fixed during both loading and dumping. When so constructed special apparatus must be used to load the bucket while it is in motion. The traveling speed is necessarily slower and seldom exceeds 300 feet per minute.

The jig-back or two-bucket reversing tramway is similar in general operation to the bicable tramway, except that only two buckets are used, one on each track cable, and the running rope instead of traveling continuously in one direction is reversed at each trip. Thus, each bucket travels back and forth on its own track cable. This type is designed for slow or high speeds and light or heavy loads and is sometimes operated without mechanical power similar to an inclined plane - the descending loaded bucket furnishing power to pull back the ascending empty. Each bucket is attached to one of two hoist ropes with a tail rope running from one bucket to the other. The jig back can also be used to haul loads up an incline, but mechanical power must be applied.

The to-and-fro or single-bucket reversing tramway resembles the jig back but has only one bucket running on a single track cable. The direction of travel of the traction rope is reversed each trip. The bucket is attached to one end of a hoist rope and gravity is depended upon for movement in one direction. The system is usually designed for comparatively light loads and moderate speeds but may be built for heavy loads. At times a tail rope is necessary.

The hoisting and transporting cableway is a to-and-fro single-carriage system but is so designed that the carriage may be stopped and held in any position on the track cable and the bucket lowered and hoisted from that point. It is designed for low or high speeds but for heavy single loads (up to 150 tons or more). The aerial tram transports but does not hoist loads, whereas the cableway does both. As mentioned, it is rather a specialized type of aerial conveyance.

The monocable (single-rope, saddle-clip) and the bicable are the two types used most for haulage under conditions required at sand and gravel plants. Either type is essentially an endless-rope haulage system in which the graded roadway, rails, and ties of the surface system are replaced by track cables supported on towers.

The great advantage of the aerial tram over other systems is that it requires no prepared roadway on the ground. This avoids interference with surface operations along the right-of-way and traffic at intersections with highways. This advantage is emphasized in country of rough topography and in the crossing of rivers or other bodies of water. By locating a tower on either or both banks of a river the aerial tram carries its load across on suspended cables, where a surface system would require a bridge. If the stream is navigable the surface haulage problem is further complicated by the necessity of a swing or lift bridge to permit shipping to pass. With the aerial tram haulage is simplified by increasing the height of the towers on the river bank so that the buckets pass high enough to avoid water-borne traffic. In rough topography the aerial tram proceeds in a direct line between the loading and discharge points and the haulage distance is therefore a minimum. Since the tram can negotiate favorable or adverse grades any difference in elevation between terminals causes no operating difficulty. If the grade is against the loads more power must be expended, of course, but this would be necessary with any other system of haulage between the two points. To keep within permissible haulage grades over rough topography the surface, systems, on the other hand, must either resort to heavy grading in the roadway, requiring deep cuts and high fills, or, when the difference in elevation between terminals is too great for permissible grades on a straight line, to a longer curved roadway or possibly switchbacks, all of which require more or less grading.

Like barge and truck loading systems the aerial tram may be used for either short intraplant service or for long interplant haulage. However, barge and truck systems are more flexible in that they can transfer material from one or more points to one or more other points due to their independence of a single roadway. The aerial tram, on the other hand, is confined to transfer of material from one fixed point to another, although buckets may be loaded or discharged from more than one point along the line.

The aerial-tram system compares favorably with other types of haulage in installation cost over level topography and ordinarily has considerable advantage in this respect in rugged country. Any type of power may be used, including steam or internal-combustion engines, electric motor, or line shaft. Modern installations usually employ the electric motor. A technical discussion of the application and electrical hook-up for different types of motors, as applied to aerial trams, will be found on page 419, volume 133 (1932) of Engineering and Mining Journal.

When loads are transported over level surface or adverse grades by aerial tram, power must be expended as with any other type of haulage. When the material is lowered down steep grades, as often occurs in mountainous country, the tram may be operated as a gravity plane in which the descending loads return the empty buckets without mechanical power except that of the brake drum. With some such installations electric motors are installed for use when no loads are descending. When the tram is in operation the motor acts as a brake and generates electric power for use elsewhere. Sometimes both favorable and adverse grades are necessary. The driving motor may then be used part time to operate the line and part time as a power generator. In long lines it is customary to install a starting motor in addition to the regular operating drive. The duty of the starting motor is to overcome the initial inertia of the buckets and haulage rope and start them moving. When once in motion the starting motor is cut out and the regular drive operates the line.

Structural Limitations

The possibilities and advantages of the aerial tram for haulage in rugged, mountainous, or otherwise inaccessible country have long been recognized by metal miners, but its successful competition with other systems in the simpler forms of plant haulage has progressed slowly. So far there have been few applications to sand and gravel exploitation, although recent improvements in design and construction have solved many earlier operating problems.

As previously stated the aerial tram is essentially an endless-rope haulage system in which the carrying vehicles are transported through the air instead of on the ground. To accomplish this the rails and ties of the surface system are replaced by track cables and supporting towers.

In the bicable type there are two track cables, one of larger diameter carrying the loaded buckets in one direction and the other of smaller section carrying the returning empty buckets. The track cables are anchored securely at one end and stretched over metal saddles on the towers to a tension station at the other end. In the tension tower the cables are attached to heavy weights which provide the power tension to enable the cable to support a load of similar weight suspended between towers and the weight of the traveling buckets. Tension towers may be placed at either the loading or discharge terminal on short hauls or in the middle or at other convenient intermediate points for long hauls. When the tension tower is placed in the line the buckets are transferred from one cable to the next over steel rails and rail points, thus permitting continuous operation without stopping. Track cables are made in various designs. In some the core is a round strand of wire surrounded by interlocking wires, some or all of which are specially shaped for two reasons: (1) To present a smooth surface, free from impact, for the passage of the carrier wheels, and (2) to prevent exterior broken wires from protruding or unwinding and thus avoid entanglement or derailment of the carrier. In some installations a lang-lay haulage rope is used for track cable. The locked-coil cable is more expensive but presents less friction to the wheels of the bucket carriage and usually has a longer life. Haulage rope is often used for short hauls or for temporary construction.

Track cables pass over the towers and are supported in cast-iron or steel saddles which are grooved to fit the underside of the cable but offer no obstruction to the passage of bucket wheels. The cable rests in the saddle groove and is free to move longitudinally in either direction. This is necessary, as constant shifting of the load due to the traveling buckets causes constant change in the arc of sag between towers and hence continual movement of the track cable in the saddle grooves.

A third, or haulage, rope, usually of lang-lay construction, transmits motion to the buckets. This rope is ordinarily driven by a bullwheel operating on a vertical shaft and equipped with multiple finger grips or leather lining to provide driving friction. The haulage rope may be below or above the track cables, depending upon the design of the tram. The buckets are attached to it by friction grips. These grips may be operated manually by an attendant at the loading station, or they may be so constructed as to clamp automatically to the haulage rope as the bucket is pushed onto the line. In either case the bucket is detached automatically at the discharge terminal. The bucket, in passing from one cable to another at an intermediate tension station , remains fastened to the haulage rope as it passes over the rail connection between track cables.

In the monocable type the traction or haulage rope of the bicable system is eliminated, and the heavy track cables themselves provide motion. They pass over a series of sheaves at each tower arranged so as to minimize the arc of sag at that point. No intermediate tension stations are required, but the proper tension is applied to the cables at one or both of the terminal stations. The tower sheaves are so designed that the bucket carriage passes over them without releasing the friction grip in the saddle clips. At the dumping terminal it is unnecessary to detach the bucket as must be done in a bicable system. The incoming cable is led over a sheave, at the terminal and directly alongside is placed the graded end of a steel rail. As the bucket approaches the terminal the carriage sheaves ride the rail and lift the saddle clips clear of the cable, thus detaching the bucket and transferring it to the rail where it is moved manually to the dumping position.

Aerial-tram towers may be either of wood or steel, depending upon local economic factors. They are short or long depending upon the contour of the surface, which also determines whether their spacing along the line shall be at equal distances or irregular intervals. If spaced at short distances less tension is needed on the track cables, but the cost for tower construction is greater. Close spacing also causes less deflection in the track cables as the loaded buckets approach and leave a tower saddle. This results in less wear on the cable and hence longer life. Tower spacing must be designed carefully to provide an equable balance between initial installation cost and maintenance expense. Where the tram passes over highways, rivers, or other bodies of water the towers on either side are usually higher than at other points to provide clearance for highway or water-borne traffic. Where the tram spans rivers or valleys the distance from bank to bank or hill to hill often far exceeds the customary tower spacing. Long spans require greater tension on the track cables and necessitate greater sag in the suspended cables. This increased sag may require increased tower height, even though there is no traffic interference in the span. Single spans of this nature are frequently necessary in mountainous country and have been operated successfully over distances approaching a mile in length. In crossing highways it is customary to suspend a steel-wire network below the tram buckets as a safety precaution against dropping of material on the highway.

Although longitudinal movement of the track cables is provided for in the saddle grooves no provision is made for movement of the cables at right angles to the direction of bucket travel. The weight of the cable and buckets normally prevents the cable from moving out of the groove. Under abnormal conditions, however, such as excessive or unequal loading, the track cable may be lifted vertically out of the saddle. In that case, if the towers are not alined carefully, the cable when it drops back will settle either on the tower cap inside the saddle or outside the end of the cap. In either case the next bucket passing is apt to catch at the tower or drag on the ground and cause trouble. For this reason aerial trams are built without horizontal curves in most installations, and great care is necessary to assure that the tower saddles are in exact alinement. Modern design, however, has devised mechanical appliances which permit considerable curvature in the line, but a special tower must be erected at the point of the curve to accomplish this.

The usual method is to pass the track cables around a deflecting sheave of large diameter and bypass the buckets around the curve from one tangent to the other by means of steel rails on the curve tower. This requires an attendant at the station to reclamp the buckets to the haulage rope after the latter has passed its deflecting sheave, as it is difficult to devise a clamp that will retain its grip while passing around such a sheave.

The aerial tram can be constructed to transport material over any sort of surface and for any reasonable distance. It may be utilized to transfer material from one building to another a few hundred feet apart, each building functioning as a terminal with an unsupported span between buildings, or it may function as a haulage unit many miles long. For example, the Peru Mining & Smelting Co. employs an aerial tram to haul ore from its mines to the seaboard more than 30 miles away. However, these long trams are not built in one self-contained operating unit. They are usually divided into independent units 1 to 6 or 7 miles long, each unit delivering to the next one either through a transfer tower containing storage bins or by passing the buckets from one section to the next. Each section has its individual power drive. This division into sections also permits an angular change of direction with minimum difficulty.

The average sand and gravel operator, however, would not be as interested in a long haul as in a single self-contained unit handling his material from pit to plant or from plant to transportation terminal.

Aerial-tram buckets vary in design. Some are carried on axles similar to surface cars and travel above the track cables. With such buckets two track cables carry the loads and two return the empties. The wheels on the ends of the axles are grooved to fit the cables, and the haulage rope is above the track cables. In the more commonly used design a bucket is hung in a structural-steel carriage or frame supported from a pin in a 2- or 4-wheel trolley running on a single track cable. The bucket hangs below the track cables, and tha haulage rope is attached to a clamp on the supporting frame. Each tower also carries idler rolls which carry the sag of the haulage rope between buckets. Both designs have their advocates and critics, but the latter seems to be more popular, judging from the number of installations.

Aerial trams are not suited to handling material direct from excavators. Being tied to a fixed loading point the tram cannot follow the excavator. Some other form of haulage which is flexible must intervene between excavator and tram.

Aerial-tram buckets are comparatively small units and consequently are not suitable for handling large lumps or boulders. The efficiency of a tram depends largely on its ability to keep a large number of small-unit carriers in constant motion. This necessitates quick loading and dumping. Material containing a large percentage of boulders presents obvious difficulties to quick loading, hence the tram operates more efficiently with crushed or small material that can be loaded quickly and smoothly through chutes from bins.

Bucket sizes vary with the individual installation, depending upon the capacity desired, the physical character of the material handled, the rope speed of the tramway, and the spacing of the buckets on the line. Trams have been operated successfully with buckets having capacities ranging from 500 pounds to 4 tons each. A mine in Germany using autotrucks to take material from the excavator has designed the truck body so that it can be picked up by a sling on the aerial tram and used as a tram bucket. This arrangement avoids the necessity for crushing machinery and loading bins between the excavator and the tram.

Table 74 has been compiled from articles in the technical press to show the characteristics and range of aerial tramways.

As stated previously, aerial-tram buckets are usually loaded from bins through chutes. Loading bins serving conveyor belts cannot be made portable owing to mechanical difficulties involved in varying the length of the haul. Loading stations can be inserted, however, at various points along the tram, hence loading is not confined to one point.

Tram buckets can be dumped by hand or by automatic devices at any point along the line. Usually buckets are dumped into receiving bins, but sometimes the tram is used to deliver to ground storage or waste dumps. The dumping point may then be at a tower or between two towers as desired. In the latter case automatic dumping devices are fastened to the track cables at the desired point.

Auxiliary Machinery

If the aerial tram is considered as a system consisting of towers, track cables, haulage rope, traveling buckets, and power supply, the only auxiliary machinery required is that for loading and dumping the buckets. This usually involves bins at both loading and discharge terminals. Loading bins are filled by some other haulage unit, such as a conveyor belt, power scraper, or cableway excavator. If the excavated material contains large pieces or boulders it is customary to install a crusher over the loading bin or at some point in the flow of material preceding the bin. In installations such as that mentioned in Germany loading bins and crushers are eliminated, but there are few examples of this type of installation.

Since the tram is tied to a stationary loading point material must be brought to it by other haulage systems; thus they may be considered as service equipment.

At the discharge terminal a bin is commonly provided over which the buckets may be operated by hand or by the haulage rope and dumped at any point desired either manually or automatically. Usually buckets must be detached from the haulage rope at this point as at curve stations. Discharge bins may serve as temporary storage to feed the washing or treatment plant or may be merely intermediate points in the haul supplied by one section of the aerial tram and loading another section or some other type of haulage such as railway cars or autotrucks.

TABLE 74. - Aerial-tram installations

Tram no.	Haulage distance, feet	Bucket capacity, pounds	Number of buckets	Bucket spacing Ft.	Bucket spacing Sec.	Haulage speed, ft. per min.	Track cable diameter, inches load	Track cable diameter, inches empty	Haulage rope diameter inches
1 1/	31,050	896	---	200	29	420	1	1	---
2 2/	1,200	1,000	5	---	--	---	-	-	---
3 3/	26,400	1,200	100	---	--	---	1 3/8	1	5/8
4a 4/	10,162	2,000	---	200	--	500	1 1/2	7/8	5/8
4b	5,150	3,800	---	228	27.4	500	1 3/4	1	3/4
5 5/	2,500	2,000	16	320	48	400	1 1/2	7/8	5/8
6 6/	1,655	2,400	21	324	--	350	1 3/4	1 1/4	3/4
7 7/	6,671	3,000	88	157	--	450	1 5/8	1	1
8 8/	3,000	---	---	---	--	---	-	-	---
9 9/	6,600	4,000	80	---	--	---	-	-	3/4
10a 10/	740	8,000	2	---	--	---	2 1/4	2 1/4	7/8 & 3/4
10b	1,060	8,000	2	---	--	---	2 1/4	2 1/4	7/8 & 3/4
11a 11/	21,921	---	---	---	--	550	1 3/8	1 1/8	3/4
11b	9,000	---	---	---	--	500	1 1/8	1	3/4
11c	35,265	---	---	---	--	550	1 3/8	1 1/8	3/4
11d	30,560	---	---	---	--	550	1 3/8	1 1/8	3/4
11e	31,413	---	---	---	--	550	1 3/8	1 1/8	3/4
11f	16,008	---	---	---	--	550	1 3/8	1 1/8	3/4
11g	17,778	---	---	---	--	550	1 3/8	1 1/8	3/4

Tram no.	Longest clear span, feet	Angles No.	Angles Degrees	Difference in elevation of terminal, feet	Capacity, tons per hour	Power, HP Start	Power, HP Run
1 1/	2,925	1	40	0	60		70
2 2/	---	0		-220	15		5
3 3/	---	-		---	25		50
4a 4/	---	2	(144 (170	+33	150		50
4b	---	0		+8	250		30
5 5/	---	0		0	75		--
6 6/	450	0		+110	135		30
7 7/		0		(+420 (-215	260	30	10
8 8/	---	0		-200	180		25
9 9/	750	0		0	---		60
10a 10/	740	0		-357	150		0
10a	1,200	0		-642	150		0
11a 11/	---			+8	12.5		40
11b	---			-559	50		30
11c	---			-1,202	15	40	75
11d	---			-2,063	15	40	75
11e	---			+780	15	20	40
11f	---			(-5,321	15	75	
11g	---				15	75	

1/Indian Copper Corporation, Min. Mag., vol. 50, no. 6, June 1934, p. 340. 2/Clearwater Lime Products Co., Orofino, Ida., Pit & Quarry, vol. 19, no. 12, Mar. 12, 1930, p. 67. 3/Northwest Magnesite Co., Pit & Quarry, vol. 19, no. 12, Mar. 12, 1930, p. 51. 4/Penn Dixie Cement Corporation, Pit & Quarry, vol. 27, no. 4, October 1934, p. 35. 5/Rock Products, vol. 36, no. 8, August 1933, p. 27. 6/Ross Island Sand & Gravel Co., Rock Products, vol. 33, no. 2, Jan. 18, 1930, p. 25. 7/Superior Portland Cement Corporation, Pit & Quarry, vol. 20, no. 2, Apr. 23, 1930, p. 41. 8/Basalt Rock Co., Napa, Calif., Pit & Quarry, vol. 18, no. 9, July 31, 1929, p. 35. 9/U.S. Gypsum Co., Alabaster, Mich., Rock Products, vol. 33, no. 19, Sept. 13, 1930, p. 105. 10/Olympic Portland Cement Co., Pit & Quarry, vol. 19, no. 12, Mar. 12, 1930, p. 41. 11/Peru Mining & Smelting Co., Eng. & Min. Jour., vol. 134, 1933, p. 287.

Capacity

The capacity of an aerial tram depends upon the size of the individual buckets, their spacing on the haulage rope, their speed of travel, and the method of loading them.

For the same haulage distance the capacity can be increased as follows: (1) By substituting an equal number of larger buckets, (2) by using more buckets of the same size but decreasing the distance between them, or (3) by increasing the traveling speed.

For an increased haulage distance the same capacity may be maintained by simply increasing the number of buckets on the line, the spacing and speed remaining constant. The length of haul therefore affects capacity only if a limited number of buckets is available.

Within certain mechanical limits any or all three of these methods can be used to increase capacity.

Often the size of the buckets can be increased without changing the size of the track cables, but before this is attempted care must be exercised to see that the increased concentration of load between any two towers does not cause the track cable to be lifted clear of the saddle on the next tower. This is apt to occur when the tram extends over a long, level or fairly uniform ascending or descending route or where loading and discharge terminals are at higher elevations than intervening towers.

The same care must be exercised in increasing the number of buckets by closer spacing, even though the bucket size is not increased. Decreasing the spacing may cause two loads between towers where formerly only one was necessary. This doubles the traveling load on that particular span, increases the sag in the track cables, and tends to lift the latter from the saddles in succeeding towers.

In either instance a reduction in the tension on the track cables may keep them in their saddles, if sufficient clearance above ground is available, but in so doing the sag is increased between all towers, and the deflection at each tower is increased as the buckets pass over it, thus increasing cable wear due to movement in the saddle and to flexure. Usually this evil is corrected by increasing the height of the tower from which the track cable is being lifted.

Increasing capacity either by larger buckets or more buckets of the same size may require greater tension on the track cables, and this may cause them to lift off the saddles on low towers. Either method may also necessitate replacing the track cables with others of larger diameter, which in turn may require a redesign of the trolley wheels on the bucket carriage.

Capacity can often be increased by increasing haulage speed, but this entails careful study of the driving mechanism, particularly with respect to friction on the drive wheel and the setting of the bucket clamps. Increased speed requires greater driving tension and tighter bucket clamps.

The capacity of an aerial tram is directly affected by the efficiency of the loading device. When it is loaded from bins the chute and gate must be quick-acting, and the material must flow easily and smoothly through the chute to give greatest efficiency. If the material contains boulders or clay rapid loading is difficult owing to blocking in the chutes. When material of this type is hauled by aerial tram it is customary to enlarge the loading station so that surplus empty buckets are always available for loading from more than one chute. A timing device is also helpful so that the operator may send the loaded buckets away at regular intervals. Timing devices may be run by gears from the main shaft and ring a gong at the proper interval, or markers may be placed on the haulage rope at the required distance. The former device is more satisfactory and more generally used.

Owing to the variation in length, traveling speed, and other local conditions it is impracticable to tabulate aerial-tram capacities based on either bucket size or speed. Capacity must be computed for each installation according to local conditions. When all factors are known capacity can be computed accurately if no correction is necessary to account for partly-filled buckets. Since tram buckets usually are not loaded automatically but by manpower, calculation of capacity must take into account the personal equation, as with other types of equipment, hence working capacity is always less than theoretical capacity. The difference will depend upon the efficiency of the loading device and the loader.

The theoretical capacity of an aerial tram may be computed in numerous ways; the following is probably as simple as any.

Let

C = theoretical capacity of the tram, in tons per hour;
W = rated load capacity of each bucket, in pounds;
N = number of buckets delivered per hour

Then

$$C = \frac{WN}{2,000}$$

Let

v = rated volume of each bucket, in cubic feet, and
m = average weight of a cubic foot of the material carried.

Then

$$W = vm$$

Let

t = time interval between buckets, in seconds;
d = distance interval between buckets, in feet;
s = speed of the haulage rope, in feet per minute.

Then

$$N = \frac{3,600}{t} = \frac{60s}{d}$$

and by combination in the original formulas

$$C = \frac{WN}{2,000} = \frac{vmN}{2,000} = \frac{9vm}{5t} = \frac{3vms}{100d}$$

The formula used will depend upon the form in which the known data are presented.

Example 14

An aerial tram is assumed to have been installed with the following operating conditions, and it is desired to know its capacity when handling sand and gravel.

The speed of the haulage rope (s) is 450 feet per minute.

Each bucket has a capacity (v) of 9 cubic feet.

The buckets are spaced (d) on the line at 180 feet.

Sand and gravel ranges in weight from 2,900 to 3,240 pounds per cubic yard, or 107.4 to 120 pounds per cubic foot.

Therefore m will range from 107.4 to 120 pounds.

Then in the formula,

$$C = \frac{3vms}{100d},$$

C will range from

$$\frac{3x9x107.4x450}{100x180}$$

or 72.5 tons per hour at the minimum value of m to

$$\frac{3x9x120x450}{100x180}$$

or 81.0 tons per hour at the maximum value of m.

In example 14 it was assumed that a tram was already set up, and the calculation was made to determine merely its capacity when handling sand and gravel. Field calculations usually involve a different approach to capacity problems. The operator wants to know the details of a tram that will deliver a certain quantity of material from one point to another in a given time. Example 15 illustrates this approach.

Example 15

An operator wishes to erect a tram that will deliver his material, which averages 110 pounds per cubic foot (m), from pit to plant at the rate of 120 tons per hour. Then

$$120 \text{ tons per hour} = \frac{120x2,000}{60} = 4,000 \text{ pounds per minute.}$$

With a rope speed of 450 feet per minute (s) and buckets spaced to arrive at 30-second intervals (t), the distance spacing, (d), is

$$\frac{450}{2} = 225 \text{ feet.}$$

Individual bucket loads are

$$\frac{4000}{2} = 2,000 \text{ pounds.}$$

This size of the buckets (v) is

$$\frac{2,000}{110} = 18.2 \text{ cubic feet,}$$

but to prevent spillage, 20-cubic foot buckets are selected.

Then by formula

$$C = \frac{3vms}{100d} = \frac{3x18.2x110x450}{100x225} = 120 \text{ tons per hour.}$$

The theoretical capacity is

$$\frac{3x20x110x450}{100x225} = 132 \text{ tons per hour.}$$

With the same rope speed but a 20-second delivery interval the distance spacing would be

$$\frac{450}{.3} = 150 \text{ feet.}$$

Individual bucketloads would be

$$\frac{4,000}{3} = 1,333 \text{ pounds,}$$

and each bucket would require

$$\frac{1,333}{110} = 12.1 \text{ cubic feet}$$

with, say, 15-cubic foot buckets installed.

The working capacity would then be

$$C = \frac{3x12.1x110x450}{100x150} = 120 \text{ tons per hour,}$$

and theoretical capacity would be

$$C = \frac{3x15x110x450}{100x150} = 148 \text{ tons per hour.}$$

Similarly, various other bucket sizes and line spacings could be selected.

Power Calculations

The power required to operate an aerial tram may have as its source the force of gravity, a mechanical motor, or both.

Power is consumed in (1) moving the buckets and their loads horizontally, (2) elevating buckets and loads vertically, (3) overcoming resistance offered by line friction, (4) overcoming resistance offered by friction in terminal machinery, and (5) overcoming inertia in starting and accelerating the buckets to line speed.

If the design of the tramway is such that the material transported must also be elevated, then the installed motor must have enough power to do all these things.

If the tramway is designed to lower material, then the force of gravity may be sufficient to overcome all power-consuming factors and may even develop a surplus for use elsewhere, but with low grades gravity is seldom enough, and additional power is necessary even though the loads move downhill.

Various authorities have compiled formulas for computing the power required to operate an aerial tram or the power developed by it10/. The object

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10/ Peele, Robt., Mining Engineer's Handbook: John Wiley & Sons, 2d Ed., New York, 1927, pp. 1745 - 1788.

being to calculate the size of the power unit necessary for economical operation. Formulas differ owing to different methods of calculation or different mechanical data. Consequently, the power requirement found by using one formula may differ from that found by another, even though the basic data are the same. This difference may or may not be serious, but usually analysis will show that the variation is due to the use of different coefficients of frictional resistance.

Frictional resistance is encountered in aerial trams in various forms. In a bicable system the bucket-carrier wheels are subjected to rolling resistance or friction as they travel over the track cables. This will vary with the type of construction used in the track cables and the tension to which they are subjected. Locked steel track cables with smooth surfaces present less rolling friction than haulage or hoist ropes used as track cables. Taut cables offer less friction than slack cables. The bucket-carrier wheels also offer journal friction in their axles, and the traction rope encounters frictional resistance in being dragged over supporting idler sheaves.

Monocable tramways must overcome rolling friction as the rope passes over the tower sheaves, which also offer journal friction.

Both types of trams must overcome frictional resistance in moving the terminal machinery. Terminal resistance or friction will vary with the tension applied to the traction rope. Wire ropes are designed to operate at what is called "working tension", which is 16 to 20 percent of their breaking strength, depending on whether a factor of safety of 6 or 5 is used. Terminal tension can be computed on the basis of the breaking strength of the traction rope. In the bicable system the traction rope performs one function only - bucket movement - hence the tension need be no more than that required to move the loaded buckets and provide sufficient friction at the drive wheel. In the monocable system the running rope performs two functions; it moves the buckets and supports them between towers. It must therefore be under greater tension and sometimes produces greater terminal friction than is required by the traction rope of a bicable system. (However, the increased power cost due to higher terminal friction may be offset by the lower maintenance cost of the simpler monocable system.)

Since plain, roller, or ball bearings may be used in both carriage and tower sheaves and in some terminal machinery, obviously the coefficient of friction will vary considerably according to the construction details.

It is customary to combine the effect of rolling and journal friction in carriage wheels and tower sheaves and represent both by one coefficient designated as "line friction." The coefficient for line friction as given by various authorities ranges from 0.0067 $\left(\frac{1}{150}\right)$[11] to 0.03 $\left(\frac{3}{100}\right)$[12]. The lower figure represents the best modern equipment, including roller bearings.

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[11] Information given the author by manufacturers.

[12] Lay, Douglas, Design of Aerial Tramways: Eng. and Min. Jour., vol. 109, June 19, 1920, p. 1359.

The higher figure probably represents plain bearings and may even include a surplus to cover terminal friction.

The coefficient for terminal friction as given by various authorities ranges from 0.00556 ($\frac{1}{180}$)[13] to 0.015 ($\frac{1}{67}$)[14].

For preliminary estimates the coefficient of line friction may be taken as ranging from 0.007 for antifriction bearings to 0.02 for plain bearings and from 0.005 to 0.015 for terminal bearings.

In using any coefficient, however, allowance must always be made for inefficient lubrication, which may totally destroy the value of any coefficient and the accuracy of computations.

In view of the preceding discussion the horsepower required to operate an aerial tram hauling loads up-grade or the power developed when the loads move down-grade may be computed from the following formulas.

$$HP = \frac{PV}{33{,}000}$$

in which

HP = horsepower, applied or produced;
P = net pull on the traction rope, pounds;
V = speed of the traction rope, feet per minute.
$P = T - S + p$

in which

T = pull on the taut side, in pounds;
S = pull on the slack side, in pounds;
p = pull due to terminal friction.
$T = WH \pm FWL$[15]
$S = wH \pm FwL$[16]

in which

W = weight per foot on the loaded side, in pounds;
w = weight per foot on the empty side, in pounds;
H = difference in elevation between terminals, in feet;
F = coefficient of line friction;
L = horizontal length of the tram, in feet.

$$W = \frac{g+e}{d} + r$$

$$w = \frac{e}{d} + r$$

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[13] Information given the author by manufacturers.

[14] Peele, Robt., Mining Engineer's Handbook: John Wiley & Sons, 2d ed. New York, 1927, p. 1773. Marks, Lionel S., Mechanical Engineer's Handbook: McGraw-Hill Book Co., 3d ed. New York, 1930, p. 1419.

[15] Sign of the last term is "plus" when the rope is pulled up-grade and "minus" when it is let down.

[16] See footnote 15.

in which

g[17/] = weight of the load in one bucket, in pounds;
e = weight of one empty bucket and carriage, in pounds;
d = distance between buckets, in feet;
r = weight of 1 foot of traction rope, in pounds.
$L = \sqrt{D^2 - H^2}$,

in which D = the inclined length of the tram, in feet

$$p = 0.8\ Bf,$$

in which

B = breaking strength of the traction rope, in pounds, and
f = coefficient of terminal friction.

Combining and substituting the above in one formula:

For loads up-grade,

$$HP = \frac{[(WH + FWL) - (wH - FwL) + 0.8\ Bf \quad V]}{33{,}000}$$

For loads down-grade,

$$HP = \frac{[(WH - FWL) - (wH + FwL) - 0.8\ Bf \quad V]}{33{,}000}$$

The preceding formulas assume that the size and weight of the traction rope are known, but none attempts to compute what that size and weight should be.

The size of the traction rope depends upon the total tension to which it will be subjected. This total tension must not be confused with rope pull, although it is based on it. Traction ropes are subject to the same laws as any band drive and may therefore be calculated from the power formulas for band drives.

The pull transmitted to the line by the traction rope depends upon the friction between the rope and the surface of the bull wheel or driving drum. The driving wheel or drum may be a steel sheave with unlined rope groove; the groove may be lined with wood, rubber, or leather; or the sheave may be fitted with self-gripping fingers. In any case, the pull it can exert depends on friction, which, in turn, depends on the tension applied to the slack side of the sheave. By analytical mechanics the relation between the tension on taut and slack sides to prevent slipping is expressed in the following formula:

17/ In calculating the starting pull g is usually doubled.

$$T - S_t e^{f\pi n}$$

in which

T = taut-side tension, in pounds;
S_t = slack-side tension, in pounds;
e = base of Napierian logarithms = 2.71828;
π = 3.1416,
f = coefficient of friction between rope and sheave;
n = number of half laps (180°) on the sheave.

$$T-S = S_t(e^{f\pi m}-1)$$

from which the slack-side tension necessary to prevent slipping may be computed and the total rope tension and size of traction rope.

For convenience the values of f and $e^{f\pi n}$ for the most common conditions are given in table 75.

TABLE 75. - Values of f and $e^{f\pi n}$

Number of half laps (n)	Condition of surfaces	Type of sheave					
		Plain, unlined		Wood-lined		Leather-or rubber-lined	
		f	$e^{f\pi n}$	f	$e^{f\pi n}$	f	$e^{f\pi n}$
1	Wet	0.08	1.3	0.17	1.7	0.40	3.5
1	Dry[1/]	.17	1.7	.23	2.1	.49	4.7
3	Wet	.08	2.2	.17	4.9	.40	43.4
3	Dry[1/]	.17	4.9	.23	9.2	.49	106.2

[1/] Figures for wet surfaces are customarily used.

Bucket and carriage weights are not standardized but vary with the design of different manufacturers. There is no definite relationship between weight and size. Table 76 shows the approximate range in size and weight.

TABLE 76. - Approximate weight of aerial-tram bucket and carriage

Size, cubic feet		Weight, pounds	
From	To	From	To
4	6	300	375
7	15	400	800
16	25	550	1,000

Example 16

The conditions are assumed to be the same as those used in example 15. The average weight of the sand and gravel was 110 pounds per cubic foot, the size of the buckets was 20 cubic feet, the distance between buckets was 225 feet, the speed of the traction rope was 450 feet per minute, and the load in one bucket was 2,000 pounds.

Additional assumptions are as follows: The inclined distance between terminals is 3,000 feet, the discharge terminal is 100 feet above the loading terminal, plain bearings are used with a line-friction coefficient of 0.02, plain bearings are used with a terminal-friction coefficient of 0.015, and bucket and carriage are assumed to weigh 800 pounds.

The size of the traction rope depends upon the pull required, the calculation of which involves the weight and therefore the size of the traction rope. Calculation must then be a matter of "cut and try."

With a bicable system and the loads pulled up-grade the weight per foot, if the rope weight is neglected, becomes:

$$W = \frac{g+e}{d} = \frac{2,000 + 800}{225} = 12.4 \text{ pounds}$$

$$w = \frac{e}{d} = \frac{800}{225} = 3.55 \text{ pounds.}$$

The inclined distance between terminals is given as 3,000 feet.

$$L = \sqrt{D^2 - H^2} = \sqrt{3,000^2 - 100^2} = 2,998.33 \text{ feet.}$$

The ratio of difference in elevation or lift to length is so small in this instance that it is disregarded, and 3,000 feet is taken as the horizontal length of the tram. With a steep incline, however, this ratio will be appreciable and must be accounted for in the calculation.

Substitution of the values of W and L in the equations for the pull on the taut and slack sides gives

$$\begin{array}{lr}
 & \underline{\text{Pounds}} \\
T = WH + FWL = (12.4 \times 100) + (0.02 \times 12.4 \times 3,000) = & 1,984 \\
S = wH - FwL = (3.55 \times 100) - (0.02 \times 3.55 \times 3,000) = & \underline{142} \\
T - S = & 1,842
\end{array}$$

If a plain, unlined sheave is used to drive with three half laps then the slack-side tension is

$$T\text{-}S = S_t(e^{f\pi n-1})$$
$$1,842 = S_t(2.2-1)$$
$$S_t = \frac{1,842}{1.2} = 1,535 \text{ pounds.}$$

Since the taut-side pull was 1,984 pounds and the tension required on the slack side is 1,535 pounds, then by addition the total tension in the rope is 3,519 pounds; thus

3,519 x 5 = 17,595 pounds = 8.8 tons breaking stress.

From manufacturers' catalogs a 6 by 7 cast-steel rope 9/16 inch in diameter has a breaking stress of 9.4 tons and weighs 0.48 pound per foot, whereas the same type of rope ½ inch in diameter has a breaking stress of 7.5 tons. Since the breaking stress of the ½-inch rope is less than that required, the 9/16-inch size is chosen. Recalculating,

$$W = \frac{2,000 + 800}{225} + 0.48 = 12.88 \text{ pounds}$$

$$w = \frac{800}{225} + 0.48 = 4.03 \text{ pounds.}$$

$$\begin{array}{lllr} & & & \underline{\text{Pounds}} \\ T = & (12.88 \times 100) & + (0.02 \times 12.88 \times 3,000) = & 2,061 \\ S = & (4.03 \times 100) & - (0.02 \times 4.03 \times 3,000) = & \underline{161} \\ T\text{-}S = & & & 1,900 \end{array}$$

$$1,900 = S_t(2.2-1), \; S_t = \frac{1,900}{1.2} = 1,583$$

$$2,061 + 1,583 = 3,644 \text{ pounds total tension.}$$

$$3,644 \times 5 = 18,220 \text{ pounds} = 9.1 \text{ tons breaking stress.}$$

This is below the tabulated strength of the 9/16-inch rope selected.

$$p = 0.8 \; Bf = 0.8 \times 18,800 \times 0.015 = 226 \text{ pounds}$$

$$P = T\text{-}S + p = 2,061 - 161 + 219 = 2,126 \text{ pounds.}$$

$$HP = \frac{PV}{33,000} = \frac{2,126 \times 450}{33,000} = 29$$

The motor rating would be 25 percent higher, making the required motor 36.25 HP, the nearest commercial size to which would be 40 HP.

Incidentally, since S or the line pull on the slack side is 161 pounds and the slack-side tension necessary is 1,583 pounds, the tension weight necessary to provide this tension is

$$2\,(1,583 - 161) = 2,844 \text{ pounds.}$$

Assuming the same conditions except that the best antifriction bearings on both line and terminal are used, the trial calculation for rope size would be

$$\begin{array}{lllr} & & & \underline{\text{Pounds}} \\ T = & (12.4 \times 100) & + (0.007 \times 12.4 \times 3,000) = & 1500 \\ S = & (3.55 \times 100) & - (0.007 \times 3.55 \times 3,000) = & \underline{280} \\ T - S = & & & 1220 \end{array}$$

If a leather-lined sheave wheel is used with one half lap, then

$$1,220 = S_t(3.5 - 1)$$

$$S_t = \frac{1,220}{2.5} = 488$$

1,500 + 488 = 1,988 pounds total rope tension
1,988 x 5 = 9,940 pounds = 5 tons breaking stress.

This requires a 7/16-inch, 6 by 7, cast-steel rope weighing 0.29 pound per foot and having a breaking stress of 5.8 tons.

Recalculating,

$$W = \frac{2{,}000 + 800}{225} + 0.29 = 12.69 \text{ pounds.}$$

$$w = \frac{800}{225} + 0.29 = 3.84 \text{ pounds.}$$

$$\begin{array}{lr}
 & \underline{\text{Pounds}} \\
T = (12.7 \times 100) + (0.007 \times 12.7 \times 3{,}000) = & 1{,}537 \\
S = (3.8 \times 100) \;-\; (0.007 \times 3.8 \times 3{,}000) \;= & \underline{300} \\
T\text{-}S = & 1{,}237
\end{array}$$

$$1{,}237 = S_t(3.5 - 1)$$

$$S_t = \frac{1{,}237}{2.5} = 495 \text{ pounds.}$$

1,537 + 495 = 2,032 pounds total rope tension.

2,032 x 5 = 10,160 pounds = 5.08 tons, which is below the tabulated strength of the 7/16-inch rope selected.

$$p = 0.8\,Bf = 0.8 \times 11{,}600 \times 0.005 = 46.4 \text{ pounds.}$$

$$P = 1{,}537 - 300 + 46.4 = 1{,}283 \text{ pounds}$$

$$HP = \frac{1{,}283 \times 450}{33{,}000} = 17.5$$

If 25 percent is added the motor rating is 22 HP. The tension weight in this case would be 2(495 - 300) = 390 pounds.

If a monocable system is used instead of the bicable the traction rope must have strength enough to support the loads over the longest span without excessive sag or deflection in the line. The permissible sag may be limited by local surface profile or economical tower height. Line sag or deflection depends upon the tension applied to the traction rope, the length and inclination of the span, the weight of the rope, the weight of the buckets and their loads, and the distance between buckets. Tramway design permits a maximum sag of 2½ to 10 percent of the span, but it usually does not exceed 5 percent. Sag or deflection must be calculated for each position of the bucket or buckets on each span and involves formulas for catenary or parabolic curves and higher mathematics.[18] For final tram design the deflection should be accurately calculated from these formulas by an engineer familiar with this class of construction.

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[18] Carstarphen, F. C., A Simple Method of Computing Deflections of a Cable Span Carrying Multiple Loads Evenly Spaced: Trans. Am. Soc. Civ. Eng., Paper 1,454, vol. 83, 1919-20, pp. 1383-1408.

Calculation of the power required for a monocable system involves the tension required in the traction rope rather than the sag. However, tensio depends upon rope weight, span length, and weight of the loads carried, all of which are involved in computing sag. Therefore, if the maximum sag is assumed the formula for sag may be used to determine the tension required and hence the size of rope necessary.

The tension in an unloaded rope may be found from the following formula:

$$Te = \frac{w\ s^2}{8\ d},$$

and the tension due to a single load at the center of a span, the ends of which are at the same elevation, by

$$Tg = \frac{P\ s}{4\ d}$$

The total tension on the rope is then Te + Tg. This will be modified when the ends of the span are at unequal elevations. In these formulas,

Te = horizontal tension due to empty rope, in pounds;
Tg = horizontal tension due to a single load, in pounds;
W = weight of the rope per foot, in pounds;
s = length of the span, in feet;
P = gross weight of one bucket and load, in pounds;
d = sag or deflection at the center of the span, in feet.

The greater deflection will be caused by the single load in the center of the span. If a ratio of sag to span of 5 percent and a maximum span of 200 feet are assumed, the tension due to load is

$$Tg = \frac{P\ s}{4\ d} = \frac{2,800 \times 200}{4 \times 10} = 14,000 \text{ pounds.}$$

This may for the moment be assumed as the working tension of the traction rope. Its breaking strength will then be

$$14,000 \times 5 = 70,000 \text{ pounds,}$$

or 35 tons. Reference to manufacturers' catalogs shows that a 6 by 19 special plow-steel rope of 1-inch diameter has a breaking strength of 42 tons and weighs 1.6 pounds per foot.

Recalculation gives

$$\begin{array}{lll} & & \underline{\text{Pounds}} \\ Te = \frac{w\ s^2}{8\ d} = \frac{1.6x(200)^2}{8x10} & = & 800 \\ Tg = \frac{P\ s}{4\ d} = \frac{2,800x200}{4x10} & = & \underline{14,000} \\ Te + Tg = & & 14,800 \end{array}$$

$$14,800 \times 5 = 74,000 \text{ pounds}$$

or 37 tons, as the breaking strength necessary, indicating a 1-inch rope.

Substitution in the power formulas gives

$$W = \frac{g+e}{d} + r = \frac{2,000 + 800}{225} + 1.6 = 14.0 \text{ pounds},$$

$$w = \frac{e}{d} + r = \frac{800}{225} + 1.6 = 5.1 \text{ pounds}$$

$$\begin{aligned} T &= (14.0 \times 100) + (0.02 \times 14.0 \times 3,000) = 2,240 \text{ pounds}, \\ S &= (5.1 \times 100) - (0.02 \times 5.1 \times 3,000) = 204 \text{ pounds}, \\ p &= 0.8\ Bf = 0.8 \times 84,000 \times 0.015 = 1,008 \text{ pounds}, \\ P &= T - S + p = 2,240 - 204 + 1,008 = 3,044 \text{ pounds}, \end{aligned}$$

$$HP = \frac{PV}{33,000} = \frac{3,044 \times 450}{33,000} = 41.5.$$

The motor rating would be 25 percent higher, making the required motor 50 HP.

With antifriction bearings the theoretical HP required would be 22.2 and the motor rating, 28 HP.

Then

$$T - S = 2,240 - 204 = 2,036$$

With a driving sheave equipped with steel jaws and having a leverage ratio of 1 to 3 the value of $e^{f\pi n}$ will be 2.2, or the same as a plain sheave with 3 half laps. Then

$$\begin{aligned} 2,036 &= S_t(2.2-1), \\ S_t &= 1,700 \text{ pounds}. \end{aligned}$$

The tension necessary on the slack side to prevent slipping is then 1,700 pounds. However, the tension necessary to provide the designed line sag was 14,800 pounds, hence slack-side tension for driving may be disregarded.

The tension weight required will be twice the necessary tension, or 14,800 x 2 = 29,600 pounds.

Aerial-Tram Design

This paper might be greatly extended by discussing aerial-tram design, including the proper heights of towers to provide clearance for loaded buckets, the tension to be applied to obtain the required deflection and driving friction, and other technical problems involved. Since, however, the solution of a single problem for a specific installation could not be applied to another because of local differences and because such a solution involves higher mathematics, the author feels that such discussion is beyond the scope of this circular. For such information the reader is referred to the references cited, to technical handbooks, and to the engineering staffs of manufacturers of aerial trams.